TRANSACTIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

VOLUME 36

THIRTY-SIXTH ANNUAL MEETING
PHILADELPHIA, PA., JANUARY 28-31, 1930

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1930

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Committee on Code for Minimum Requirements for the Heating and Ventilation of Buildings: Perry West, *Chairman;* C. A. Booth, W. H. Driscoll, H. P. Gant, E. Vernon Hill and A. C. Willard.

Committee on Code for Testing and Rating Concealed Gravity Type Radiation: R. N. Trane, *Chairman;* E. H. Beling, W. F. Goodnow, Hugo Hutzel, A. P. Kratz, E. J. Vermere and O. G. Wendel.

Committee on Code for Testing and Rating Steam Unit Heaters: D. E. French, *Chairman;* H. W. Page, L. C. Soule and W. A. Rowe; representing Industrial Unit Heater Association: G. E. Otis, Chairman; O. K. Dyer and J. H. Shrock.

Committee for Interpreting Code for Rating Low Pressure Heating Boilers: L. A. Harding, *Chairman;* R. V. Frost and F. C. Houghten.

Committee on Meetings Program: H. H. Angus, *Chairman;* A. J. Nesbitt and F. B. Rowley.

Committee on Natural Ventilation: J. E. Emswiler, *Chairman;* W. C. Randall and Andrew Vogel.

Committee on Rating and Testing Low Pressure Heating Boilers: R. V. Frost, *Chairman;* H. M. Hart, F. C. Houghten, H. F. Hutzel, and Raymond Newcomb.

Committee on Revision of Constitution and By-Laws: Thornton Lewis, *Chairman;* W. T. Jones and F. R. Still.

Committee on Rules of Award for F. Paul Anderson Medal: F. C. McIntosh, *Chairman;* J. D. Cassell and Roswell Farnham.

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1930

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TRANSACTIONS

of

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

No. 847

THIRTY-SIXTH ANNUAL MEETING, 1930

THE 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, held at the Benjamin Franklin Hotel, Philadelphia, January 28 to 31, 1930, was one of the most successful meetings of the Society. While the First International Heating and Ventilating Exposition, held in conjunction with the Annual Meeting, helped to bring registration to the impressive total of 1,093, more than double the attendance at any preceding meeting, it is also true that the previous record for registration was held by Philadelphia for nine years.

During the seven business and technical sessions of the four-day meeting the three outstanding accomplishments were : (1) Acceptance of the Code for Rating Heating Boilers using Solid Fuel; (2) Adoption of a Standard Code for Testing and Rating Steam Unit Heaters; (3) Approval of New Regulations for the Research Laboratory.

Among the other subjects discussed were papers on: heat transmission, salvaging of steam, infiltration, piping for hot water heating systems, temperature regulation, air conditioning and fan furnace heating.

Pres. Thornton Lewis opened the meeting, and greetings in behalf of the Philadelphia Chapter were extended by J. C. Cassell, honorary chairman of the Committee on Arrangements. The response was made by Vice-Pres. L. A. Harding, after which numerous communications from prominent Society members were read.

R. A. Wolff, chairman of the Tellers of Election, presented the following report:

Report of Tellers of Election

The Board of Tellers of Election appointed by Pres. Thornton Lewis, have

examined the ballots of members eligible to vote and take pleasure in rendering the following report:

For President—L. A. Harding, 674.

For First Vice-President—W. H. Carrier, 672.

For Second Vice-President—F. B. Rowley, 675.

For Treasurer—C. W. Farrar, 671.

For Members of the Council (for 3 years)—R. H. Carpenter, 673: J. D. Cassell, 672; Alfred Kellogg, 672; John Howatt, 675.

Members of Committee on Research—C. V. Haynes, 660; W. T. Jones, 666; J. F. McIntire, 665; F. N. Speller, 663; A. C. Willard, 662.

Scattering votes were recorded for various other members.

TELLERS OF ELECTION,
R. A. WOLFF,
J. V. CAVILEER,
W. R. MURPHY.

The President's Report was then read by Mr. Lewis, as follows:

Report of President

First, I desire to thank the officers, the Presidents of the Chapters, the Chairmen and all the members of our various committees. All that this administration has accomplish is due to their loyalty to the Society and their untiring efforts in its behalf.

Particular mention should be made of the work of W. T. Jones, Chairman of the Finance Committee; Samuel R. Lewis, Chairman of the Guide Publication Committee; our Vice-President, L. A. Harding, who with his associates, R. V. Frost and F. C. Houghten, has worked so faithfully on the Boiler Rating Code; Prof. A. C. Willard, Chairman of the Publication Committee.

Cecil W. Farrar's work as Chairman of the Increase of Membership Committee speaks for itself. Through his efforts a number of members have been added to our roll. We now have the largest membership in the history of the SOCIETY.

H. P. Gant, Chairman of the International Heating and Ventilating Exposition Committee, has, I am sure you will agree, done a splendid job and the Exposition will be a credit to the managers of the International Exposition Co. and to the Society.

Last, but by no means least, I desire to mention the effective, efficient and untiring work of our Secretary, A. V. Hutchinson. He never forgets, never overlooks anything which the President should do. He is everywhere and anywhere he is needed or that a job is to be done for the Society. I fear our members who have not worked closely with our New York headquarters, do not fully appreciate the versatility and general ability of Mr. Hutchinson. May I commend Mr. Harding to his good ministrations.

Your President has visited all the Chapters excepting only the one on the Pacific Coast.

There are no recommendations to the Society that I wish to leave on record; perhaps they never would be read, anyway. The policies inaugurated by this administration, if good, will continue; if not, they deserve to meet a timely end.

May I, however, summarize the chief thoughts which have actuated your President:

1st—To encourage the establishment of a Society Endowment Fund.

2nd—To further co-operation with other organizations, such as the U. S. Public Health Service, the *Heating and Piping Contractors National Association*, the *National Warm Air Heating Association*, the *American Oil Burner Association*, the *National Association of Fan Manufacturers*, the *Boiler and Radiator Institute*, the *Industrial Unit Heater Manufacturers Association* and many others.

3rd—To urge all members to interest the younger men of our industry in becoming members of our Society. They need the Society today, and we need them, for they will be the engineers, executives and manufacturers of tomorrow.

Conscious of the honor you have done me, the support you have given me, and the generosity of thought bestowed in judging my actions, may I show a slight bit of my appreciation by making this the shortest Presidential report on record.

Respectfully submitted,

THORNTON LEWIS, *President.*

Two reports, that of the Secretary and Council, were then read by A. V. Hutchinson, Secretary, as follows:

Report of Secretary

With the expansion of the Society's activities and the necessarily increased staff, 1929 has been productive in the completion of a number of Society projects, which have been under discussion and preparation for a considerable time. An unusual burden has been successfully handled in the installation of a new system of records in the Headquarters Office, and 50 per cent more membership applications were handled than in the year previous.

The numerous details connected with the change in publication method of THE JOURNAL of the Society during two-thirds of the year have been successfully handled through the splendid co-operation of the publishers and the staff.

Since the last meeting, the Code of Minimum Requirements for the Heating and Ventilation of Buildings has been printed and mailed, as well as Volumes 33 and 34 of the TRANSACTIONS for 1927 and 1928, a Code for Testing and Rating Steam Unit Heaters has been prepared and distributed to the membership, as well as the Report of the Continuing Committee on Testing and Rating Low Pressure Heating Boilers.

The Secretary has had the opportunity of visiting every Chapter except the Pacific Northwest group and has greatly enjoyed the personal contact with the members.

The interest in Chapter activities is particularly keen at present and inquiries have come from Baltimore, Montreal, Los Angeles and San Francisco about the possibility of establishing Chapters in those cities.

The rapid advance of the heating and ventilating industry, which is so graphically demonstrated in the Exposition, has stimulated interest in many quarters, and it is especially noticeable in the number of colleges that are establishing heating and ventilating classes and also doing research in heating and ventilating problems.

The necessity for greater outside contacts by the SOCIETY is evident if the organization is to maintain its leadership.

Respectfully submitted,

A. V. HUTCHINSON, *Secretary.*

Report of the Council

Since the organization meeting, January 31, 1929, in Chicago, the Council has held seven meetings and passed on many important questions affecting the operation of the Society with the result that the organization is found at a high point in its career in respect to strength of membership and financial stability.

A number of special committees were appointed on specific problems, among which were the Committee to Establish a Society Endowment Fund, Committees to Prepare Code for Testing Unit Heaters, Concealed Radiators, Unit Ventilators, and a Committee on Natural Ventilation.

One of the special problems of the Council is the increase of membership and the retaining of the interest of the older members.

A special committee, headed by C. W. Farrar, was assigned to interest the right kind of men who would qualify to join the Society, and another committee, headed by F. D. Mensing, was assigned the difficult problem of communicating with delinquent members.

The results in both cases have been most gratifying and the members are indebted to these committees for their splendid work.

Matters of special importance to the membership were the change in the monthly JOURNAL of the Society and its incorporation as a part of the magazine, *Heating Piping & Air Conditioning,* starting in May, 1929; arrangements for the Heating and Ventilating Exposition; and the participation of the Society in the World Engineering Congress at Tokio, where a paper was presented by W. H. Carrier and prepared by a committee of members of the Society under the direction of Mr. Carrier.

One of the important actions taken by the Council was in reference to the issuance of publicity material by the U. S. Public Health Service, resulting in a conference by a committee of Society members and Surgeon General Cummings, resulting in the preparation of a bulletin in co-operation with members of the Public Health Service, giving the present status of ventilation standards for application in schoolroom heating and ventilating.

The Council accepted the invitation of the Minnesota Chapter to hold the Semi-Annual Meeting 1930 in Minneapolis, and through its committee handled routine assignments of budget preparation, election of members, and publication of Transactions and technical meeting programs.

Through the individual reports of various Council Committees, the Society members will find that the activities of the Society have been of greater scope this year than in any other year, that the assets of the Society have increased over the previous year, and that the gain in the Society's General Fund through profits of the Publication Department have been substantially increased.

The opportunity for the Society to perform greater public service is evidenced by the interest of the public in its activities and the substantial increase in membership resulting this year indicates that the Society's service to the individual member is recognized.

Respectfully submitted,

L. A. HARDING, *Chairman,* Executive Committee.

President Lewis paid tribute to the work of C. W. Farrar, as chairman of the Committee on Increase of Membership, stating that for some time there had been no increase in the membership of the Society, and the Committee had increased the membership about 10½ per cent. It was pointed out that the Committee secured a total of 363 applications, 75 of whom had affiliated during the first two months, 166 having affiliated before the first of the year, so that although 1930 was started with 1,995 members, a gain of only seven over January 1, 1929, the result of Mr. Farrar's work was just beginning to show; and from January 1, 1930 to March 1, 1930, 75 members had affiliated, making the membership 2,070, the largest in the history of the society. There were 122 application still to be acted upon by the Membership Committee.

Report of Committee to Investigate Cause of Loss of Membership in the Society

Your committee on Loss of Membership would report that in compliance with the request of your president, Thornton Lewis, it made investigations into the reasons underlying the loss of membership due to accumulated back dues, and finally summed the reasons as follows:

1. Loss of interest in the Society due to change in business and like causes.
2. A feeling of resentment against the Society due to a misunderstanding of the Constitution and By-Laws of the Society.
3. Financial difficulties due to unemployment.

The approach in all cases was made to the delinquent members by means of friends, more particularly the proposers or seconders to the petitions of the delinquent members when this was possible. The financial returns of this campaign have been most satisfactory.

This can be best exemplified in its relation to its work in the Research Laboratory. The Laboratory, as you know, gets 40 per cent of the membership dues. The returns to the Research Laboratory for the past three years are as follows:

1927	$ 13,513.54
1928	12,722.96
1929 (paid and due)	209,494.52

Respectfully submitted,

January 28, 1930. F. D. MENSING.

President Lewis expressed appreciation for the excellent work of W. T. Jones, chairman of the Finance Committee, who then presented his report as follows:

Report of Finance Committee

W. T. Jones explained the difficulty of obtaining a complete audit between the time of closing the books on January 1, and the day of the report of his Committee on January 28. He then submitted in abstract the report of the Finance Committee, bringing out the important features of the operating statements of Society and Publication activities as summarized in the Report of the Certified Public Accountant.

President Lewis added that the agreement with the publishers of the JOURNAL of the Society had resulted in a profit to the Society of over $7,000 on eight months' operation after deducting the cost of members' subscriptions.

Report of Committee on Research

Four meetings of the Committee on Research were held during the past year. At the first meeting in Chicago during the last Annual Meeting of the Society, plans for the work of the year were discussed.

Seven members of the Committee attended a meeting in Pittsburgh during the time of the Open House at the Research Laboratory, November 4, 1929. The work of the several technical advisory committees and the work carried on at the Society's Laboratory and in the cooperating university laboratories was discussed and plans perfected for the future.

A third meeting of the Committee in Buffalo on December 20 was attended by 11 members. A number of matters of vital importance to the future progress of the Laboratory were discussed at this meeting. A committee made up of D. S. Boyden, Chairman, Philip Drinker, R. S. Franklin, and assisted by W. T. Jones was appointed to revise and bring up to date the regulations for the guidance of the Committee on Research in the operation of the Research Laboratory. The proposed regulations after approval by the Council are to be presented to the Society at the annual meeting for adoption.

It was decided at this meeting that the Research Laboratory should engage in research and testing for associations and also for individuals and firms or corporations under certain restrictions as provided by the new regulations, who wish to finance the work involved. In this connection, it was agreed that the Laboratory would, when requested, carry on certain research and testing desired by the *Heating and Piping Contractors National Association.*

It was also decided at the Buffalo meeting of the Committee to recommend to the Council of the Society the appointment of a full-time paid executive secretary to the Research Committee in order to relieve the chairman of many of the routine burdens of his office and to make a more concerted effort to obtain funds for the research activity of the Society and give more effective publicity to the work of the Laboratory.

This Laboratory has 12 research projects under investigation or under consideration for investigation. The work on these problems is outlined by the following technical advisory committees:

Temperature, Humidity and Air Motion: W. H. Carrier, *Chairman;* O. W. Armspach, S. C. Bloom, C. A. Bulkeley, Philip Drinker, W. A. Rowe, Perry West.

Infiltration: G. L. Larson, *Chairman;* A. W. King, A. P. Kratz, L. B. Lent, W. C. Randall.

Heat Transmission: A. P. Kratz, *Chairman;* D. R. Brewster, F. B. Rowley, C. G. Segeler, Perry West.

Pipe Sizes for Heating Systems: H. M. Hart, *Chairman;* S. E. Dibble, R. S. Franklin, F. E. Giesecke, C. V. Haynes, R. C. Morgan.

Heating and Ventilation in Its Relation to Health and Comfort: Philip Drinker, *Chairman;* John Aeberly, O. W. Armspach, C. P. Yaglou.

Radiation: R. A. Wolff, *Chairman;* C. W. Brabbée, R. V. Frost, A. C. Willard, C. H. B. Hotchkiss, J. F. McIntire, F. D. Mensing.

Garage Ventilation: E. K. Campbell, *Chairman;* W. H. Carrier, E. B. Langenberg, Thornton Lewis.

Air Cleaning Devices: F. B. Rowley, *Chairman;* H. E. Birkholz, Albert Buenger, E. V. Hill, H. C. Murphy.

Atmospheric Dust and Smoke: E. B. Langenberg, *Chairman;* E. V. Hill, S. R. Lewis, H. C. Murphy.

Oil Burning Devices: L. E. Seeley, *Chairman;* James Breese, Jr., G. S. Meikle, H. L. Tapp.

Effect of Wind and Weather Conditions on the Heating Loads: R. S. Franklin, *Chairman;* W. L. Fleisher, J. F. Hale, E. B. Langenberg, S. R. Lewis, F. R. Still, A. C. Willard.

Testing and Rating Unit Heaters: D. E. French, *Chairman;* O. K. Dyer, G. E. Otis, H. W. Page, W. A. Rowe, J. H. Schrock, L. C. Soule.

Annual reports of the Technical Advisory Committees submitted to the Chairman of the Committee on Research are appended.

A large part of the work of the Laboratory is now carried on in ten universities and colleges in accordance with contracts or cooperative agreements. The institutions cooperating with the Laboratory during the past year and the projects on which they worked are as follows:

Armour Institute of Technology—Measurement of air flow through registers and grilles.

Carnegie Institute of Technology—Pipe sizes for steam heating.

Harvard University—Vital characteristics of the atmosphere.

University of Kentucky—Study of methods of testing unit heaters.

University of Minnesota—Heat transmission through building construction; and the efficiency of air cleaning devices.

Purdue University—Standardization of method of testing radiators.

Agricultural and Mechanical College of Texas—Pipe sizes for hot water heating.

Washington University—Garage ventilation in relation to carbon monoxide and fire hazards.

Wisconsin University—The air leakage through walls.

Yale University—Standardization of methods of testing oil burners.

In each of these institutions with the exception of Carnegie Institute of Technology, the research is under the supervision of the university which furnishes all overhead for the work in addition to one-half of the salaries of the investigators carrying on the work.

Carnegie Institute of Technology cooperates to the extent of furnishing laboratory space, heat, power and other facilities for the pipe size investigation which is carried on by the Research Laboratory staff.

Besides the pipe size study which is carried on at Carnegie Institute of Technology, the Laboratory staff is working on the following problems at the Research Laboratory in the Pittsburgh Experiment Station of the U. S. Bureau of Mines:

(1) Heat and moisture losses from the human body and their relation to air conditioning problems.

(2) Air leakage through a stucco and a hollow tile brick veneer wall.

(3) Change in the conductivity of concrete with time.

(4) The absorption of solar radiation by a wall or roof.

During the past year the study of heat loss through walls with the Nicholls heat flow meter was discontinued and another phase of the heat transmission problem was taken up—namely, the study of surface transfer coefficients for various types of sur-

faces met with in building construction. In order to better understand heat transfer for these surface conditions a study was also undertaken of wind velocity gradients near a wall. The velocity gradients for a wind parallel to the wall surfaces has been completed. Apparatus for measuring the surface heat transfer was constructed and some data on the subject collected.

As a result of the cooperative work carried on by the Research Laboratory ten papers were presented to the Society during the year—five at the last summer meeting and five at the present annual meeting:

Capacity of Radiator Supply Branches for One and Two-Pipe Systems, by F. C. Houghten, M. E. O'Connell and Carl Gutberlet.

Determining Dust Quantities in Air by Prof. F. B. Rowley and John Beal.

Heat and Air Volume Output of Unit Heaters, by L. S. O'Bannon.

Over-all Heat Transmission Coefficients as Determined by Test and by Calculation, by F. B. Rowley, A. B. Algren and J. L. Blackshaw.

Pipe Sizes for Hot Water Heating Systems, by Prof. F. E. Giesecke and E. G. Smith.

Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle, and other Characteristics of the Absorbing surface, by F. C. Houghten and Carl Gutberlet.

Air Infiltration Through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz.

Measurement of the Flow of Air Through Registers and Grilles, by L. S. O'Bannon.

Pipe Sizes for Hot Water Heating Systems, by Elmer Smith.

Surface Transmission Coefficients, by F. B. Rowley, A. B. Algren and J. L. Blackshaw.

The results of the Laboratory's study of relation of atmospheric condition to physiological reactions of man are of great interest to the medical and physiological profession and the results of the Laboratory's study of heat and moisture loss from adults at rest were published in the April, 1929, *American Journal of Physiology*. A second paper on heat and moisture loss from men working has since been submitted for publication.

Besides the results published during the year, the Laboratory has data on air leakage through walls, change of conductivity of concrete with time, and the capacity of pipe for the returns of steam heating systems.

The finances of the Laboratory are in good shape. Receipts for the year were slightly greater than expenditures so that a little was again added to the surplus. Contributions pledged are reported as follows:

Contributions to the A. S. H. V. E. Research Fund in 1929

Name	*Amount*
Aerofin Corp.	$ 100.00
Air Filter Association	600.00
American Blower Co.	1,000.00
American District Steam Co.	200.00
American Steam Pump Co.	50.00
Armstrong Cork and Ins. Co.	150.00
Barnes and Jones	200.00
Bishop and Babcock Sales Co.	50.00
Blaney, Chas. A.	100.00
Buffalo Forge Co.	250.00
Carrier Engineering Corp.	300.00
Celotex Co.	100.00
Engineering Publications, Inc.	100.00
C. A. Dunham Co.	600.00
Heating-Piping Contractors Nat. Assn.	500.00
Hoffman Specialty Co.	250.00
Ilg Electric Ventilating Co.	200.00
Illinois Engineering Co.	300.00
Johnson Service Co.	500.00
Kewanee Boiler Co.	250.00
McAlear Mfg. Co.	300.00

Modine Mfg. Co.	$ 200.00
Nash Engineering Co.	335.00
National Lumber Mfg. Assn.	322.87
National Regulator Co.	250.00
Nesbitt, John J., Co.	250.00
N. Y. Blower Co.	150.00
Ontario Chapter A.S.H.V.E.	132.00
Phila. Chapter A.S.H.V.E.	100.00
Powers Regulator Co.	200.00
B. F. Sturtevant Company	750.00
Taylor Instrument Co.	50.00
Trane Co.	200.00
U. S. Gypsum Co.	50.00
Ventilating Contractors Employers Assn. of Chicago	400.00
Walworth Co.	100.00
Warren Webster & Co.	1,000.00
York Heating and Ventilating Corp.	400.00

Report of the Certified Public Accountant

January 21, 1930.

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
29 WEST 39TH STREET
NEW YORK CITY

Gentlemen:

Pursuant to your request I made an examination of the books of account and records of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, New York City, for the year ended December 31, 1929, and submit herewith the following exhibits and comments:

Exhibit
"A" Balance Sheet, December 31, 1929.
 Schedule No. 1—Marketable Securities.
"B" Statement of Income and Expenses of the Society for the Year Ended December 31, 1929.
 Schedule No. 5—Salaries.
"C" Statement of Income and Expenses of the Publications for the Year Ended December 31, 1929.
 Schedule No. 2—Cost of JOURNAL.
 Schedule No. 3—Cost of GUIDE.
 Schedule No. 4—TRANSACTIONS.
 Schedule No. 5—Salaries.
"D" Comparison of Budget, Society Activities.
"E" Comparison of Budget, Publications.

CASH

The Cash on Deposit was verified by direct communication with the following banks and reconcilement of the amount reported to me with the balances shown by the books of the Society:

Bankers Trust Company, 42nd Street and Fifth Avenue, New York City	$5,146.74
Bank of America, N. A., 40th Street and Madison Avenue, New York City	1,437.43
Total	$6,584.17

A count was also made of the Petty Cash on hand and found in agreement with the General Ledger Interest Fund account.

MARKETABLE SECURITIES

A schedule showing a list of investments made by the Society in Marketable Securities kept with the Bankers Trust Company for safekeeping is included as part of this report. The shrinkage in market value of the securities as of the close of business, December 31, 1929, of $4,554.03 is not reflected in the attached Balance Sheet.

ACCOUNTS RECEIVABLE

A trial balance was taken of the Dues Receivable from members as of December 31, 1929, and classified as to the year invoiced. A summary of the dues is shown below:

1929 Unpaid Dues	$ 7,605.15
1928 " "	2,822.50
1927 " "	1,089.53
Dues Receivable	$11,517.18

Dues amounting to $447.09, which had been prepaid during the past year, are shown on the attached Balance Sheet as a deferred income. After reviewing in detail all available data concerning the collectibility of 1929 unpaid dues a reserve in the amount of $3,602.08 was provided. This, together with the balance of the Reserve carried forward from the previous year in my opinion will be ample to cover probable losses in the realization of all Dues Receivable.

The membership of the Society on December 31, 1929, comprised the following:

Members	1,424
Associates	405
Juniors	156
Students	1
Honorary	2
	1,988

Trial balances of all other Accounts Receivable were also taken and the proper reserve to cover accounts doubtful of collection provided.

INVENTORIES

The Inventory of TRANSACTIONS taken on December 31, 1929, was submitted for my verification and is scheduled below:

YEAR	VOLUME	NUMBER	PRICE	AMOUNT
1922	28	154	$1.18⅜	$ 182.34
1923	29	125	1.97½	246.87
1924	30	222	1.38¼	306.92
1925	31	181	.96⅛	174.06
1926	32	117	1.26⅘	148.36
1927	33	349	1.32⅛	461.38
				$1,519.93

TRANSACTIONS covering the Years 1928, Volume 34, and 1929, Volume 35, were not published up to the close of business, December 31, 1929, therefore reserves in the amount of $2,500.00 and $3,000.00, respectively, have been provided to cover the future cost of producing those volumes.

A verification of the Inventories and Deferred Charges to future operations was made either by computation or actual count.

ACCOUNTS PAYABLE

A list was compiled of all invoices remaining unpaid on December 31, 1929, dating prior to January 1, 1930, for the purpose of determining all Accounts Payable and a liability therefor was set up in the sum of $17,768.71. This sum includes charges from Horn-Shafer Co. of $14,410.00 for costs covering the printing of THE GUIDE and YEAR BOOK.

DUE RESEARCH LABORATORIES

Of the dues charged to Senior and Associates 40 per cent has been reserved for the Research Laboratory in accordance with Section 5, Article 3 of the By-Laws. The sum payable to the Research Laboratory as and when the Dues Receivable will have been realized in cash is $16,465.82. In addition, there is due the Research Laboratory the sum of $4,973.50, representing that portion of the profit resulting from THE GUIDE, which has been computed as follows:

Income from Guide	$46,348.78	
Cost of Guide	33,414.04	
Gross Profit		$12,934.74

Overhead Expense:		
Amount charged to Guide Calendar Year 1928	$ 2,222.14	
Add: One-half of Increase of 1929 Expenses over 1928	1,634.46	$ 3,856.60
		$ 9,078.14
Deduct: Chapter Meeting Expenses		1,000.00
		$ 8,078.14
Bonus to Publication Management and Staff (88%)		3,104.64
Net Profit from THE GUIDE for Research Laboratory		$ 4,973.50

ACCRUED ACCOUNTS

There has been determined a bonus of $3,528.00 to be paid to the publication management and staff which has been deducted as an expense in the Statement of Income and Expenses from the Publications.

FUNDS

A resolution of council provides that there be set up a publication fund for all gross revenue derived from the JOURNAL during the period from May 1 to December 31, 1929, and in accordance therewith the sum of $10,666.70 realized has been shown as the publication fund on the attached Balance Sheet. An analysis of the funds of the Society reflecting the changes that occurred therein during the past year is shown on page 11.

Fund—December 31, 1928, per Former Report		$35,416.95
Add: Profit from Publications for the year ended December 31, 1929, from Statement of Income and Expenses	$ 4,611.11	
Deduct: Loss on Society Operations for the year ended December 31, 1929, from Statement of Income and Expenses	769.80	
Net Profit for the Year		3,841.31
Funds—December 31, 1929, per Balance Sheet		39,258.26
Less: Publication Fund		10,666.70
General Fund—December 31, 1929, per Balance Sheet		$28,591.56

Respectfully submitted,

FRANK G. TUSA,
Certified Public Accountant.

BALANCE SHEET

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, NEW YORK CITY

December 31, 1929

ASSETS

CASH			
On Deposit—General Fund	$ 6,584.17		
On Deposit—Research Fund	1,680.54		
		$ 8,264.71	
On Hand		100.00	
			$ 8,364.71
MARKETABLE SECURITIES—PER SCHEDULE			
General Fund	25,596.53		
Add: Accrued Interest	765.63		
		26,362.16	
Research Fund		3,045.61	
			29,407.77
ACCOUNTS RECEIVABLE			
Dues	11,517.18		
Less: Reserve for Doubtful	7,358.08		
		4,159.10	

Advertisements	$37,423.79			
Guides	1,255.00			
Codes	326.00			
Reprints	99.74			
		39,104.53		
Less: Reserve for Doubtful		1,000,00	38,104.53	
				42,263.63
INVENTORIES				
Transactions			1,519.93	
Transactions Paper			524.68	
Pictures			72.00	
Emblems			95.87	
Reprints and Books			98.45	
Code of Minimum Requirements			543.31	
				2,854.24
LIBRARY				
FURNITURE AND FIXTURES			5,794.50	
Less: Reserve for Depreciation			3,290.25	
				2,504.25
DEFERRED CHARGES				
PUBLICATIONS				
Mailing			1,412.34	
Prepaid Local Chapter Expenses			500.00	
				1,912.34
				$87,606.94

LIABILITIES

ACCOUNTS PAYABLE				$17,768.71
DUE RESEARCH LABORATORY				
Dues			$11,307.73	
1929 Guide Profit			4,973.50	
Interest on Securities			67.50	
				16,348.73
ACCRUED ACCOUNTS				
Bonus—Publication Management and Staff				3,528.00
RESERVE FOR TRANSACTIONS				
1928			2,500.00	
1929			3,000.00	
				5,500.00
DEFERRED INCOME				
Prepaid Dues				477.09
FUNDS				
General		$28,591.56		
Publications		10,666.70		
			39,258.26	
Research			4,726.15	
				43,984.41
				$87,606.94

NOTE "A." There were no contingent liabilities reported to me and as far as could be ascertained none existed.

NOTE "B." This Balance Sheet is subject to the comments contained in the letter attached to and forming a part of this report.

BUDGET COMPARISON—SOCIETY ACTIVITIES

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, NEW YORK CITY

For the Years Indicated Below

	Actual 1928	Actual 1929	Budget Provision	Increases Decreases
BUDGETED INCOME				
1929 Dues Income	$24,738.49	$25,906.25	$25,000.00	$ 906.25
Prior Years Dues Realized	5,072.55	4,005.12	6,000.00	1,994.88
Initiation Fees	2,499.00	3,845.00	3,000.00	845.00
Sale of Emblems and Certificate Frames	237.50	310.41	250.00	60.41
Sale of Code of Minimum Requirements	0	3,165.82	5,000.00	1,834.18
Interest Earned	1,175.14	2,002.77	1,200.00	802.77
Profit from Sale of Securities	200.00	0	0	0
TOTALS	$33,922.68	$39,235.37	$40,450.00	$ 1,214.63
BUDGETED EXPENSES				
Salary—Secretary	$ 2,750.00	$ 3,600.00	$ 3,600.00	0
Salary—Clerical	5,420.00	5,655.66	6,000.00	$ 344.34
Rent	1,000.00	1,000.00	1,000.00	0
Professional Services	360.00	450.00	400.00	50.00
Postage	1,219.84	1,643.38	1,500.00	143.38
General Printing	970.36	1,836.87	1,000.00	836.87
Yearbook	862.78	721.37	850.00	128.63
Cost of Emblems and Certificate Frames	113.31	351.52	140.00	211.52
Traveling—Secretary	1,335.90	1,121.47	1,000.00	121.47
Traveling—President	687.51	1,000.00	1,000.00	0
Traveling—Technical Secretary	0	136.20	250.00	113.80
Meetings—Annual and semi-annual	3,099.58	2,747.25	3,000.00	252.75
Local Chapter Meeting Allowance	1,000.00	1,000.00	1,000.00	0
Special Council and Committees	358.17	981.53	1,000.00	18.47
Exhibit—Power Show	64.35	95.90	100.00	4.10
Allowance for Members' Journal Subscriptions	3,966.00	2,982.00	3,000.00	18.00
Allowance for Members' Transactions Subscriptions	1,983.00	2,982.00	3,000.00	18.00
Due	60.00	60.00	60.00	0
Special Boiler Code	568.41	671.48	500.00	171.48
Code of Minimum Requirements	3,584.52	3,060.21	2,500.00	560.21
APPORTIONABLE EXPENSES				
Rent	1,577.85	1,282.82	1,340.00	57.18
Office Expenses	1,385.46	1,395.18	1,380.00	15.18
Office Supplies	430.52	200.46	480.00	279.54
Allowance for Depreciation of Furniture and Fixtures	271.80	347.67	300.00	47.67
	33,069.36	35,522.97	34,400.00	922.97
UNBUDGETED EXPENSES				
Publicity Expenses	195.73	69.00	0	69.00
Unit Heater Code	0	223.08	0	223.08
Moving Expense Technical Secretary	0	385.00	0	385.00
	195.73	677.08		677.08
	$33,265.09	$36,000.05	$34,400.00	$ 1,600.05

EXHIBIT "D"

BUDGET COMPARISON—PUBLICATIONS

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS
NEW YORK CITY

For the Years Indicated Below

	1928	1929	Budget Provision	Increases Decreases
INCOME				
JOURNALS				
Advertising	$21,802.15	$ 6,206.92	$ 5,500.00	$ 706.92
Sale of Journals	1,087.38	191.90	100.00	91.90
Editorial Contract	0	7,584.02	8,000.00	415.98
Members' Subscriptions	3,966.00	2,982.00	3,000.00	18.00
	26,855.53	16,964.84	16,600.00	364.84
TRANSACTIONS				
Members' Subscriptions	1,983.00	2,982.00	3,000.00	18.00
Copy Sales	259.61	286.17	500.00	213.83
	2,242.61	3,268.17	3,500.00	231.83
GUIDE				
Advertising	31,539.94	33,879.28	33,500.00	379.28
Sales of Guides	6,010.28	12,469.50	7,500.00	4,969.50
	37,550.22	46,348.78	41,000.00	5,348.78
SALE OF REPRINTS AND BOOKS	846.48	1,064.04	400.00	664.04
TOTALS	$67,494.84	$67,645.83	$61,500.00	$ 6,145.83
COSTS AND EXPRESS				
COST OF PUBLICATIONS				
Journal	$14,297.93	$ 4,248.37	$ 4,500.00	$ 251.63
Guide	25,385.30	33,414.04	23,600.00	9,810.04
Transactions	2,620.31	3,023.45	2,500.00	523.45
Reprints and Books	722.66	672.20	300.00	372.20
	43,026.20	41,358.06	30,900.00	10,458.06
EXPENSES				
Salary—Secretary	2,750.00	3,600.00	3,600.00	0
Salary—Clerical	4,074.00	5,253.67	4,000.00	1,253.67
Professional Services	240.00	300.00	300.00	0
Postage	337.77	328.02	400.00	71.98
Traveling—Secretary	475.46	980.25	1,000.00	19.75
APPORTIONABLE EXPENSES—40%				
Office Expenses	860.83	825.92	920.00	94.08
Office Supplies	122.98	133.64	320.00	186.36
Allowance for Depreciation of Furniture and Fixtures	181.20	231.78	200.00	31.78
	9,906.24	13,175.16	12,300.00	875.16
TOTALS	$52,932.44	$54,533.22	$43,200.00	$11,333.22

EXHIBIT "E"

Respectfully submitted,

F. G. TUSA,
Certified Public Accountant.

Report of the Technical Advisory Committee on Air Cleaning Devices

December 17, 1929

The Committee has held two meetings, one at the time of the Annual Meeting in Chicago and the other on December 11, in Chicago. At the first meeting, plans for research at the University of Minnesota were discussed, and it was decided to continue the study of the problem of determining the dust in air. Accordingly, the work which had been previously started with the A—A Dust Determinator was completed and further series of tests were made with the Hill Dust Counter.

A report of the study made on the Hill Dust Counter was given at the Semi-Annual Meeting, in a paper entitled, An Investigation of the Impingement Method of Determining the Quantity of Dust in Air. This paper was published in the A. S. H. V. E. Journal, June, 1929, Vol. 35, No. 6.

As a result of the study of the Hill Dust Counter, it was found that this Counter had one or two defects which it appeared could be easily overcome. Those defects which were causing irregularities in the results were:

1. The fact that the air velocity through the nozzle was not constant, due to the pressure created by the stroke of the pump.

2. It was found very necessary that the distance from the orifice to the glass plate be maintained exactly the same in all counters.

3. The size of the orifice affected the dust counter.

4. In order to get uniform results after the sample was taken, it was essential to place the sample under the microscope in exactly the same position and to use a microscope which was easily focused.

These changes have been discussed with Dr. Hill who is now building a new instrument which will be tried out in the laboratory.

At the meeting in Chicago on December 11 the Committee reviewed the research work and considered a request from the Technical Advisory Committee on Atmospheric Dust and Smoke to recommend changes in the Hill Counter which would insure more uniform results by different operators. As these changes were along the line of those suggested by previous research work, Dr. Hill agreed to make up a counter embodying them and send it to the laboratory for test.

The research work in connection with the Committee will be mostly with the Hill Dust Counter until the annual meeting at which time further plans will be considered.

F. B. Rowley, *Chairman;* H. E. Birkholz, Albert Buenger, E. Vernon Hill, H. C. Murphy.

Report of the Technical Advisory Committee on Testing and Rating Unit Heaters

December 19, 1929

The Unit Heater Committee planned no work during 1929, other than that done at the University of Kentucky in cooperation with the Research Laboratory of the Society, in the interest of the code for testing and rating unit heaters, and which I understand will be covered by Mr. Houghten's report on the subject.

The desirability of further work at the University of Kentucky in cooperation with the Research Laboratory in the general interest of the Unit Heater field, was discussed. A number of topics were considered, such as

(a) A formula for converting unit heater ratings at one condition of entering air temperature and steam pressure to some other condition which would properly account for change in mass velocity and which might simplify the present code method.

(b) The desirability and adaptability to all types of heaters of the method of testing at a single fan speed but against varied resistances, by comparison with the

method of testing at varied fan speeds in order to determine characteristic curves.

(c) Proof that the assumption that heat transfer from the coil for a given mass velocity of the air is proportional to the mean temperature difference between the entering air and steam pressure that holds for all coil materials and lengths of fin and character of the fin and the prime surface.

When these were referred to Professor O'Bannon with the request for an outline of tests and appropriation estimate, he felt in each case either that there was so little hope of a satisfactory solution or so little value in developing more information than was already on record, that the University of Kentucky could not recommend an appropriation.

Other work of a more routine nature, needed to confirm some of the assumptions of the Code, could be more quickly undertaken by one of the member companies, and so was not referred to the Research Laboratory.

At the last meeting of the Engineering Committee of the *Industrial Unit Heater Association,* which is composed of a representative from each of the member companies of the Association, I asked for suggestions on research work of value to the industry which might be referred to the Research Laboratory. As yet no suggestions have been forthcoming.

D. E. FRENCH, *Chairman;* O. K. DYER, G. E. OTIS, H. W. PAGE, W. A. ROWE, J. H. SHROCK and L. C. SOULE.

Report of the Technical Advisory Committee on Heat Transmission

December 12, 1929

A meeting of the committee was held June 27, 1929, at Bigwin Inn, Canada, at which time a program of tests was agreed upon, both for the Laboratory and for Professor Rowley's work at Minnesota.

The following papers have been prepared for presentation during the past year:

1. Overall Heat Transmission Coefficients as Determined by Tests and by Calculations, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. Presented at the Semi-Annual Meeting, June 28, 1929.

2. Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. Presented at the Annual Meeting, January 27-31, 1930.

3. Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet. Presented at the Annual Meeting, January 27-31, 1930.

The work has progressed following the outline adopted at the meeting of the committee of June 27. The present status can best be shown by extracts from the reports of Professor Rowley and Mr. Houghten.

The work in progress for the present year consists of hot box and hot plate tests. For the hot box test, the following schedule of walls is under way:

(a) Two rubble walls constructed 10 in. in thickness, of Blue Trenton Limestone. This material runs about 4-5 in. in thickness and will be selected in suitable widths for 10-in. walls.

(b) Six concrete block walls; two 6 in. thick, two 8 in. thick, and two 10 in. thick; the size of the blocks to be 7¾ in. in height, 15¾ in. in length, and of three-cell construction. Two types of blocks will be used: the first in which crushed limestone is used for the aggregate, and the second in which limestone is not used. These blocks will be selected from two representative block manufacturers in Minneapolis.

(c) Two concrete walls with 1-2-3½ mix with aggregate ranging from 1 in. down, one wall to be constructed with a 3-in. slump and the other with a 6-in. slump. These walls are to be poured in a vertical position in two lifts.

The hot plate tests will be as follows:

(a) Specimens 1-2-3½ mix with 3-in., 5-in. and 7-in. slump.

(b) One 1-3-5 mix with 5-in. slump.
(c) One 1-4-6 mix with 5-in. slump.
(d) One 1-1½-2½ mix with 5-in. slump.
(e) Two gypsum plaster panels.
(f) Two cement plaster panels.

These tests will be made on the 24-in. hot plate with specimens 2 in. in thickness. The concrete panels will be compared with the results as obtained by the hot box method.

The work originally outlined for determining wind velocity gradients parallel to a wall has been completed and a complete report of the work was prepared. It has appeared desirable, however, to hold up this report pending completion of the work on determination of surface coefficients of heat transfer for still and moving air.

Apparatus for the method of determining surface transmission coefficients for still and moving air as outlined by Mr. Harding has been constructed and a set of data giving a curve showing the relationship between the surface transmission coefficient and wind velocity has been completed for a neutral gray painted surface at a temperature of 90 deg with air at 70 deg, giving a 20 deg temperature difference. This work shows coefficients considerably higher than previously accepted for both still and moving air. Air velocities from a little above one mile to 23 mph have been studied. This work is just now interrupted while the temperature and heat control apparatus is being used in the calibration of several Nicholls' Heat Flow meters. As soon as the apparatus is again available—probably in about three weeks—it is our intention to develop a like curve showing the relationship of the surface transmission coefficient for the same surface and wind velocity for another temperature range. At the present time we are figuring on a temperature range of from 50 to 70 deg. If, however, a preliminary study shows that we can get a lower temperature it is our intention to use a temperature degree difference of 20 deg with as low an air temperature as we can depend on getting in our apparatus any reasonable percentage of the time.

We are continuing the tests on conductivity of the concrete slab which has been under investigation for the past two or more years. The conductivity shows little or no change. However, it will probably take another six months to insure that no change is taking place.

The status of the future program is as follows:

Further hot plate tests will be made on the various types of insulating materials in accordance with the program of June 27. These insulating materials are at present being selected by Mr. Houghten and Mr. Close.

An exhibit covering the heat transmission work is to be prepared for the Heating and Ventilating Exposition at Philadelphia. This exhibit will consist of a complete set-up of the hot plate apparatus and will occupy a part of the Research Laboratory space.

Further papers which are planned in this field will be a completion of the surface coefficients research work, and more complete results on hollow block, concrete, and tile walls. These papers will probably be ready by next spring.

A part of this work which I believe it is now safe to report is the cooperative work between the *National Lumber Manufacturers' Association*, the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, and the University of Minnesota. The *National Lumber Manufacturers' Association* has already contributed to the University for this work a sufficient amount to make about half of the tests. They are now planning on a further contribution to the University and the Society, which we believe will work out satisfactorily.

Some studies for the purpose of correlating Weather Bureau readings with arbitrarily measured velocities over actual wall surfaces is under consideration, but has not been definitely approved by the committee as yet. This will form the basis for further thought and discussion on the part of the committee.

A. P. KRATZ, *Chairman;* D. R. BREWSTER, F. B. ROWLEY, C. G. SEGELER, PERRY WEST.

Report of the Technical Advisory Committee on Infiltration

January 7, 1930

The activities of the Committee on Infiltration can be best illustrated by the two papers relating to Infiltration which will be presented by members of the Committee at the coming meeting in Philadelphia.

I refer to the papers Pressure Difference Across Windows in Relation to Wind Velocity, by J. E. Emswiler and W. C. Randall and Air Infiltration Through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz.

The work by Emswiler and Randall ties in very well with the Laboratory work on infiltration that we have been doing here.

I think our laboratory work has established quite definitely the amount of infiltration that can be expected under certain definite wind conditions in relation to brick walls, and the paper by Emswiler and Randall is a start on the actual wind conditions that may be expected in practice.

In other words, I might say that the work under the guidance of the Committee on Infiltration during the past year has established the following facts:

1. In a reasonably well constructed brick wall the air leakage is very small compared with the heat transmission.

2. Proper application of plaster either direct or on furring eliminates practically all leakage.

3. Leakage in walls is caused by faulty construction and may be anything depending on how poor the construction is.

4. The Emswiler and Randall paper has established the pressure difference across windows that may be expected in comparatively low buildings.

The work planned by the committee for the future is as follows:

1. Study of several frame walls at the University of Wisconsin.

2. Study of 1 stucco and 1 hollow tile brick veneer wall at the Research Laboratory in Pittsburgh.

The paper on brick wall construction concludes for the present at least, research work on that type of construction. Research is under way on frame wall construction and we plan to present a paper on infiltration through frame walls at the summer meeting in June.

The Committee is asking Messrs. Emswiler and Randall to continue their work on pressure differences across windows with particular reference to tall buildings.

It has also been suggested that the Committee encourage further work on weatherstrip research.

G. L. LARSON, *Chairman;* A. W. KING, A. P. KRATZ, L. B. LENT, W. C. RANDALL.

Report of Committee on Heating and Ventilation and Its Relation to Health and Comfort

December 16, 1929

The primary purpose of this committee is to study the difference between indoor and outdoor conditions, particularly the difference between an artificial indoor condition and the natural outdoor condition which it is intended to reproduce.

The first step in the study will be the observation of the degree of ionization in different atmospheres, natural and artificial. The major part of this experimental work will be carried out at the Harvard School of Public Health by means of a special instrument imported from Germany. Through the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, the experimenters have been granted funds to defray the cost of this equipment. On account of irksome delays on the part of the German manufacturer and the necessity of replacements

due to breakage en route, the apparatus is only now ready for use. We, therefore, can submit no data at this time.

If this preliminary research gives significant results, the logical sequel would be a study of methods and equipment to preduce the desired conditions indoors.

PHILIP DRINKER, *Chairman;* J. J. AEBERLY, O. W. ARMSPACH and C. P. YAGLOU.

Report of Technical Advisory Committee on Oil Burning Devices

December 30, 1929

The oil burning research program is conducted by Professor L. E. Seeley at Mason Laboratory, Sheffield Scientific School, Yale University, in co-operation with the Research Laboratory, and the *American Oil Burner Association.*

The work on oil burners was actively started about the first of October. The work to date has consisted of installing four types of small heating boilers so that complete tests might be run on any two boilers at the same time.

The oil burners are in the process of installation at the present time. There are five types of oil burners to be operated. It is the intention to rotate these burners so that each one will be operated in each boiler.

The above equipment as well as draft regulating devices and oil supply equipment has been obtained "on consignment" through the efforts of H. L. Tapp of the *American Oil Burner Association.*

The primary object of the tests which will start actively within two weeks is to develop a code for testing a combination of boiler and oil burner. This code will be developed and the tests shown as evidence to support our recommendations.

It is expected that many smaller investigations will develop, some of which may accompany the main report to be submitted in June, 1930. Certain incidental findings may be transmitted directly to Mr. Tapp who will co-operate with the respective burner companies if it seems warranted.

The appropriation for this year is not used up and it is our intention to use the remainder during the coming months. Negotiations for further appropriations will be submitted at a future date. It is our belief that we shall know more about the costs of conducting tests and that better expense estimates can be made. Our costs to date have been on installation which is in the nature of a base expense only.

It has also been agreed that the mechanical engineering department of Yale University will furnish a few pictures and examples of the work done here along the lines of heating for the Exposition in Philadelphia during January.

L. E. SEELEY, *Chairman;* JAMES BREESE, JR., G. S. MEIKLE and H. L. TAPP.

Appendix

Regulations Governing the Committee on Research of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

(Presented for adoption by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS)

January, 1930

FUNCTION

The function of the Committee on Research is the determination and dissemination of engineering knowledge pertaining to the art of heating and ventilating and the equipment and apparatus utilized by the profession.

PURPOSE

The purpose of the Committee on Research is to supervise the investigation, collection, tabulation and co-ordination of existing data and records of subjects pertaining to the art of heating

and ventilating, and when the need is sufficient to warrant research or testing, to devise a plan for such procedure; also the establishment and maintenance of a Research Laboratory, and the negotiation with universities, colleges, and other organizations provided with laboratories for co-operative research and testing work.

SECTION I

ORGANIZATION

Committee on Research

1. There shall be a standing committee known as the Committee on Research, consisting of fifteen members each serving for three years, and five retiring each year.
   (a) The Council shall nominate previous to October first of each year five members to fill the vacancies of those retiring at the next Annual Meeting.
   (b) The nominations made by the Council shall be published in the October issue of the Society's JOURNAL
   (c) Prior to December first of any year, any ten members over their own signatures, may nominate one or more additional members of the Committee on Research, and such additional nominations shall be placed on the ballot opposite the nominations made by the Council.
   (d) The election shall otherwise conform to the regulations provided for the election of officers of the Society.
   (e) Vacancies may be filled by the Council, such persons chosen by the Council to serve until a successor is elected at the next Annual Meeting.

Executive Committee

2. There shall be an Executive Committee consisting of the Chairman and two other members of the Committee on Research appointed by him immediately following the Annual Meeting.

Technical Advisory Committee

3. The Chairman of the Committee on Research shall appoint such Technical Advisory Committees and designate a Chairman of each, as the Executive Committee may deem advisable, to act in an advisory capacity to the Committee on Research and the Director of the Research Laboratory for specific projects or tests under consideration. At least one member of each Technical Advisory Committee shall be a member of the Committee on Research. The Director of the Laboratory shall be an ex-officio member of all Technical Advisory Committees.

Chairman Committee on Research

4. The Committee on Research shall at its meeting held at the time and place of the Annual Meeting of the Society each year elect by ballot vote one of their number to serve as Chairman for the ensuing year.

Vice Chairman

5. A Vice Chairman shall also be elected in the same manner and for the same period as stated for the Chairman in the preceding paragraph.

Meetings of Committee on Research

6. A meeting of the Committee on Research shall be held at the time and place of each Annual and Semi-Annual Meeting of the Society. Special meetings may be called at the discretion of the Chairman, or by a majority vote of the Executive Committee. The place and date of the special meeting to be designated by the Chairman.

Quorum

7. Seven members of the Committee on Research shall constitute a quorum.

SECTION II

DUTIES OF COMMITTEES

Committee on Research

1. The Chairman of the Committee on Research shall preside at all meetings of the Committee on Research and the Executive Committee. The Chairman of the Committee on Research shall be an ex-officio member of all Technical Advisory Committees.

2. The Vice Chairman shall possess all the powers and perform the duties of the Chairman in his absence.

3. All acts of the Committee on Research other than those specifically authorized shall be subject to review by the council.

4. All acts, findings or reports of sub-committees shall be subject to the approval of the Committee on Research.

5. The reports of all investigations conducted at the Society's Research Laboratory or resulting from co-operative agreements shall be made available to all members of the Society through publication in the Journal when authorized for publication by the Committee on Research and approved by the publication Committee.

6. All papers, findings or reports resulting from the work of the Committee on Research shall when published by the Society be headed by a picture of the Laboratory at Pittsburgh, Pa.
   (a) If the paper, finding or report is the result of work at the Pittsburgh Laboratory, the following statement shall directly follow the picture: "Research conducted by the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in co-operation with the U. S. Bureau of Mines."
   (b) If the paper, finding or report is the result of co-operative work with some other institution or laboratory, the following statement with the approval of the Director of the U. S. Bureau of Mines shall directly follow the picture: "Research conducted at (name of co-operating institution or Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS."

7. "Constitution and By-Laws; Article IX; Special Committees: *Section 1.* The Research Executive Committee, as provided for under the regulations creating the Research Laboratory, shall prepare the budget for the expenditures of the Research Laboratory for the current year, and shall pass upon and approve in writing all expenditures authorized by the Committee on Research. The payment of bills shall be authorized in the same manner as provided in Article VIII, *Section 3,* excepting that the Chairman of the Research Executive Committee shall act instead of the Chairman of the Finance Committee, and the Director of the Research Laboratory shall act instad of the Secretary of the Society, but in case of absence or disability on the part of the Treasurer the checks may be signed by the Chairman of the Committee on Research instead of the Chairman of the Finance Committee."

Executive Committee

8. The Executive Committee shall supervise all the business of the Committee on Research, including the making of the necessary contracts for rental or purchase of equipment or materials. This Committee shall select and engage a Director of the Research Laboratory, an Executive Secretary to the Committee on Research, and such assistants of the Research Laboratory as may be required, or the authority for the selection and employment of such assistants may be delegated to the Director of the Laboratory by authority of the Executive Committee in individual cases. The Committee shall determine all salaries, approve all expenses incurred, and determine the order in which the subjects shall be investigated by the Research Laboratory.

9. The Executive Committee shall not make contracts in excess of the income of the Committee on Research without the approval of the Committee on Research.

Technical Advisory Committees

10. Technical Advisory Committees shall act in an advisory capacity to the Committee on Research and the Director of the Laboratory on all subjects referred to them.
    (a) The Chairman of a Technical Advisory Committee shall be responsible for obtaining the opinion and advice from each member of his Committee concerning the assigned subject. The advice on the assigned subject as to the methods of procedure, etc., shall be transmitted to the Chairman of the Committee on Research.
    (b) In the case of a new Committtee this advice shall be furnished as promptly as conditions shall warrant in order that there shall be no unnecessary delay in starting the investigations.
    (c) Each Committee shall submit its recommendations in writing to the Committee on Research and render annually, not later than December first of each year, a written report of its activities.

SECTION III

REGULATIONS

1. That any research may be undertaken for other engineering, scientific, technical organizations, trade associations, manufacturers' associations, persons, firms, or corporations provided all work shall be related to the art or science of heating and ventilating and of such character as to meet with the approval of the Committee on Research, and that only those subjects may be investigated for which sufficient funds to pay all costs thereof have been assured.
2. That any testing of apparatus may with the written approval of the manufacturer be undertaken upon application by any engineering, scientific, technical organization, trade association, or manufacturers' association, upon request.
3. When any test of apparatus has been or is being made in accordance with the preceding paragraph, then any person, firm or corporation may have similar apparatus of his manufacture tested upon application.
4. All tests shall be conducted under the following conditions:
   (a) That the Committee on Research shall sanction each test before being made and approve the testing procedure to be used.
   (b) That the applicant shall pay all costs to the Laboratory such as: the salaries of investigators, materials, supplies, labor, expenses, and overhead.
   (c) That the applicant shall keep on deposit with the treasurer of the Society an amount equal to the estimated cost of the work for not less than two months in advance, against which amount the costs shall be charged.
   (d) The applicant shall be advised of the progress of the test and shall have the privilege of discontinuing at any time upon giving written notice, and the paying of all costs including salaries of the investigators to the end of the month following such notice.
   (e) The result of the test without comment may be authorized for publication in the JOURNAL by mutual consent of the applicant and the Committee on Research.
   (f) That the applicant shall not publish or allow to be published in whole or in part the report of the test until after publication in the JOURNAL of the Society, or by authority of the Committee on Research.
   (g) That the apparatus tested shall be removed from the Laboratory by the applicant promptly upon the completion of the test.
   (h) That the applicant shall give a bond or other security approved by the Research Executive Committee guaranteeing to the Society the full performance of the contract on the part of the applicant.

Committee: D. S. BOYDEN, *Chairman,* R. S. FRANKLIN, PHILIP DRINKER.
January 24, 1930.

DISCUSSION

D. S. BOYDEN: In presenting these regulations it should be noted that the majority of them have been in force for a good many years and in this report an attempt has been made to correlate, simplify and revise existing regulations based upon the experience of 10 years operation. Minor changes have been made with reference to the responsibility of certain committees and some revision of the restrictions regarding testing, have been included in Section III, par. 2, 3 and 4.

F. D. MENSING: I move that the recommendations submitted by the Committee on Research be approved by the Society and that all previous resolutions by this body, any rule, regulation of the Council or of the Committee on Research in conflict therewith become null and void.

H. M. HART: I second the motion.

E. V. HILL: I think that action on this matter ought to be deferred until the report is published and submitted to the entire membership for their careful

consideration. However, the point that I wish to discuss at this time concerns the use of our Laboratory for commercial testing. As I understand the rules, manufacturers can bring their apparatus and equipment to the Laboratory devoted exclusively to fundamental research, we have a commercial laboratory and if the Committee on Research sees fit it can be tested and rated. Then instead of being a laboratory devoted exclusively to fundamental research, we have a commercial laboratory. I think we would lose much of our standing and we would open the Laboratory up to the criticism which is inevitable in cases where the keen competition of manufacturers today requires a laboratory rating of their apparatus that is not favorable from a commercial angle.

PRESIDENT LEWIS: In order that the members, who may not be familiar with the previous action of the Society in regard to the Research Laboratory are brought up to date, may I correct one statement Dr. Hill has made? In 1924, five years after the establishment of the Laboratory, this Society at the Annual Meeting in New York voted to allow the Research Laboratory to undertake commercial testing, the same as is now allowed by these regulations, provided any such case had the approval of the Committee on Research. In effect, then, there is no change in principle from the original resolutions.

DR. HILL: I remember the discussion at the time. As I recall it, the general opinion of the membership was that we should not engage in commercial testing and from that date to the present we never have.

PRESIDENT LEWIS: We have not, but under the resolutions, if the Committee on Research had approved of any case we could have.

DR. HILL: I shall leave that point, and simply restate my belief that it is a dangerous proposition to encourage commercial testing.

Another point that interests me is the provision that all reports and data developed by the laboratory must be approved by the Publication Committee and released by it for publication first in the JOURNAL. It is perfectly proper to safeguard the Society and have the Publication Committee approve everything that is given publicity, but one of the things which the Laboratory has needed for many years, is more publicity. Only a very few people appreciate the wonderful work that is being done there and if we insist that papers from the Laboratory be published first in the JOURNAL, it makes the material less desirable for other publications and reduces the publicity which we could get.

I think that we ought to have a Publicity Director at the Laboratory to give the widest possible publicity to the material that is coming from the Laboratory.

MR. MENSING: If Dr. Hill will recall the 1924 resolutions in New York, I believe he will feel a good deal safer about commercial testing today than during the last six years. The field was wide open. The Research Laboratory could go as far as they wanted, but, fortunately, the research was run by brains and not be resolutions. I always prefer brains to resolutions.

I think we are perfectly safe under these new rules. Just where commercial testing stops or starts and where research begins is not well defined. Other organizations have similar problems. At one time, the *A. S. M. E.* would have hesitated to establish a standard for fittings but now the subject is not controversial. It will be the same with many things in our line. We now might consider a field one we should not go into which in future we may feel we should go into.

To get the desired publicity, the Committee has provided for an Executive Secretary, part of whose work is to handle research publicity.

S. E. DIBBLE: In considering the question of commercial testing a point that should be discussed by the entire group here, relates to our connection with the U. S. Bureau of Mines. I presume that the Committee has investigated the matter thoroughly, but I should like to learn whether it is perfectly agreeable to the present administration if we do commercial testing as indicated by Mr. Boyden's report.

MR. MENSING: If there is anybody here who can look into the future and predict future happenings with accuracy enough to answer your question, he does not belong here; he belongs in Heaven.

PROFESSOR DIBBLE: That does not quite answer the question.

L. A. HARDING: I do not know that I can give a direct answer to Professor Dibble's question, but my understanding of the arrangement under which we are operating, is, that no commercial testing can be done in the Laboratory at Pittsburgh. The idea is that commercial testing under certain restricted conditions would necessarily have to be done elsewhere if these regulations are adopted. If an association or other organization request us to do commercial testing work or to obtain the rating of apparatus, the work will go to a commercial laboratory and be supervised by our Research Laboratory staff. I doubt very much whether tests of this character would become a part of our Transactions. Occasionally there might be some new type of apparatus that was being tested on which the members would like information, but, in general, I do not imagine that such tests would be the subject of papers presented to the Society unless special request was made by members of the Society.

Many members have held for years that commercial testing was the wrong thing for this Society to do. As Chairman of this Committee I have turned down several requests this year for commercial testing, even though the existing rules and regulations permit work of this kind. The policy of preceding research committees has been to withhold approval on any work of this character which would involve the services of the Laboratory staff.

PROFESSOR DIBBLE: Mr. Harding has answered my inquiry. If the work is done outside of the U. S. Bureau of Mines, I think we will have no difficulties.

PRESIDENT LEWIS: That is my understanding of the Committee's present policy. However, the Committee may change that policy in the future should the attitude of the U. S. Bureau of Mines change.

DR. HILL: I just wanted to say, that I am in perfect accord with Mr. Mensing. I also prefer brains to resolutions. Unfortunately, we cannot always predetermine what the brains of a certain Committee or Society will be, and we can restrict their activities to a certain extent by well-considered resolutions. I thought that this was brought out, as you said, a few years ago. If commercial testing is permitted and has not been entered into, it is pretty good evidence that it is not desirable, and I still think that our rules and regulations should not permit it.

E. A. JONES: I think that the members of the Society should realize, before approving such a resolution, that it is going to lead to a demand on the part of the industry for pretty general testing. When a particular appliance is tested

by the Society, every manufacturer of similar appliances will naturally expect the Society to extend them the same privilege.

I have in mind the experience of the *American Gas Association* Laboratory which was organized a few years ago. At the present time no manufacturer of gas appliances can go into the field until he can submit a test from that laboratory.

If our Society undertakes general testing or undertakes commercial testing, we are soon going to be faced by a condition where the industry will demand the same service.

Another thing in this connection is the delegating of tests to other laboratories. I believe there would be some hesitancy on the part of certain manufacturers to send their equipment to outside laboratories rather than the Pittsburgh Laboratory for test.

President Lewis: For the benefit of everybody, I think it might be well to thoroughly understand the wording of this regulation in regard to commercial testing: "Any research may be undertaken provided that it is related to the art or science of heating and ventilating and of such character as to meet with the approval of the Committee on Research." Let us understand that wording.

Mr. Boyden: Might I add that all the regulations and revisions have been restrictions on commercial testing and if we adopt this report we carry out the policy of the Committee on Research, which has not been based on an existing resolution. The Committee's belief is that when the need is sufficient to warrant testing apparatus it may be done, but not for persons, firms and corporations. In other words, the Society when the need is sufficient can do testing for engineering, scientific, technical organizations, manufacturers' associations, etc., and only when testing is being done in accordance with the foregoing, can appliances of similar character be tested. If the Society adopts the resolution presented here, it will limit the amount of commercial testing which could be done under existing resolutions.

President Lewis: I think perhaps I might assist in clearing up this matter. The new regulations are not opening up a door that former regulations had not opened, but they put certain restrictions upon present rules. As now written any research may be conducted for anybody provided it has the approval of the Committee, but that only testing shall be done for trade associations, manufacturers' associations, etc., but not for persons, individuals and corporations, unless equipment is tested for some manufacturers' association or some trade association. We are really trying to restrict the action of the Laboratory as to commercial testing, while the previous regulations allowed the Committee on Research complete latitude in the testing of anything.

Dr. Hill: My impression is that we have had the power, had the privilege, you might say, of doing commercial testing, but the character of the Committee has been such that they have never permitted it. By throwing further restrictions around commercial testing to further safeguard it, it is anticipated that we will go into commercial testing. Am I right or wrong?

President Lewis: You are correct, I think. For instance, we were faced with the situation where a trade association this past year came to us and requested information about a new type of apparatus in order to get the desired information.

MR. HARDING: Some manufacturers have said, "We can go to one or two commercial testing laboratories, but we do not care to do that. We want a laboratory that has some recognized standing and when the apparatus is tested we want to feel that the members of the Society, other engineers and engineering organizations can accept these unbiased tests as standard." It may mean a considerable expansion of activities of the research staff.

As I view the matter, it is not destroying the function of the Research Laboratory. It is simply widening the scope of the Laboratory.

You may have to provide a Director of Testing ultimately, or something of that sort; commercial testing would not place any additional financial burden on the Society. These tests all have to be paid for and the money for tests is required to be deposited in advance.

There is always the question that is going to come up, "Are we safeguarding the individual who is not a member of some trade association?" This thought has been advanced—it may have a tendency to force small manufacturers who do not wish to join a trade association into such association in order to reap the benefits from the association's testing. That might be a good thing for the industry and, on the other hand, it might not be. It has been the history of one association, at least, that it was necessary to have an official test made on the apparatus in question before it could be employed. You had to get the approval of a certain laboratory before you could install the apparatus and obtain a proper insurance rate. If you were unable to have the test made you could not sell it. I do not believe that we are going to get into that kind of a proposition here. We are not a corporation in business for profit so that we do not come under the jurisdiction of the Federal Trade Commission.

I thought differently about this proposition and have only been recently sold on it, as a matter of fact. I believe the majority of opinions of the Committee on Research is fundamentally sound on this proposition.

PRESIDENT LEWIS: The opinion has been expressed here that as an engineering and a scientific body, conducting our own Laboratory, we should be able to furnish certain facts in regard to our industry and if a trade association comes to us and we say, "We cannot tell you," that puts our Society and our Research Laboratory in an unfavorable light. The Society ought to be the source of all scientific information for the industry and I think that is the idea of those who feel that commercial testing should be undertaken.

W. A. SWAIN: I wonder if the resolution should take into consideration the advertising man's point of view—publicity. Suppose a device has been tested by the Society and has the Society's Red Star on it and therefore it must be good. Will this seal or certificate of test bring difficulties because of its inappropriate use? Is that condition going to be taken care of in the new rules or is it just left to luck to work itself out?

C. G. SEGELER: The last speaker's words really give me an excellent opportunity to tell you what difficulties we had with that matter in the *American Gas Association* testing laboratory program. As Mr. Jones pointed out, our plan has practically forced every manufacturer who wishes to sell gas appliances to secure the laboratory seal of approval. Originally the seal of approval stood for three things: safety, quality and efficiency. Almost immediately the boomerang came back. It was extremely unsatisfactory to all manufacturers

to have any appliance have a seal on it which had the words quality or *efficiency* tied up with it in any way, because it meant that any manufacturer who had this seal immediately said in his advertising and in his sales effort: This has been tested and approved, it has a seal on it, the other man's has not, and that is what makes this product good and the other man's not good.

To remove this undesirable condition a committee studied with the result that today the seal of approval of the laboratory stands for *safety* alone.

Safety as to combustion and a minimum efficiency are required, but in no case does the seal give the manufacturer the opportunity to publish the results of the test. In fact, he does not even get them for publication.

I think that any commercial testing which your Research Laboratory would ever undertake would be forced into much the same position. Even if you first tested an appliance on the request of a trade association, say, some entirely new appliance (and perhaps a good example would be small units for house-cooling) it would not be long before another manufacturer had an appliance of that kind and there you would be in the same position of having one approved and others on the market that were not approved. I do not like to use that word approved. One tested and the others not tested. The maker of the untested one would be forced to have his appliance tested. Then the Society would be in a position of having put its name on two different appliances with two different results and that would make the Society in a sense the arbiter of the quality of those appliances.

I do not think that is a good plan. In the gas industry we allow only a certain specified minimum efficiency, for example, to be given by the manufacturer in his advertising literature. We require 75 per cent to pass and no matter whether the appliance went to 95 per cent in an actual test the manufacturer can only claim 75 as the result.

PERRY WEST: I have always been diametrically opposed to commercial testing. The safeguards that we threw around the Laboratory in the first place prohibiting commercial testing were the lifeblood of the Laboratory and brought it where it is today. Perhaps we have passed beyond that stage and we have become so well established that we do not need that safeguard quite as much as we did in the beginning.

Now that we are issuing codes of testing boilers and testing unit heaters and other pieces of apparatus there should be some authoritative place where such apparatus can be tested. Like most of the members here, I have not had the time to give this subject the same study and thought that our Council and Committee on Research have, but I am of the opinion, after hearing this discussion, that the resolution as it is presented, is perfectly safe and logical. Of course, we must realize that it may require a larger organization and the establishment of a special testing laboratory.

L. S. O'BANNON: I am not in favor of the Research Laboratory as it is organized and financed at present engaging in any kind of testing work. Creation of a separate bureau of commercial testing, should be considered, to be operated under a separate head, financed by the Society, with all personnel and equipment paid for by the association or person or corporation desiring the test. This department could call upon the resources of the Research Laboratory, or personnel for expert advice and also for the scientific apparatus and instru-

ments required for the tests. I do not know whether the committee has given thought to the idea of creating another department within the Society for that particular work or not, but I believe that that would be the better solution than to give that work to the Research Laboratory as it is at present organized.

MR. JONES: I think the professor has given us the proper solution of the commercial testing problem. The *American Gas Association* laboratory has done a good job and the manufacturers are in favor of it, but they have had a laboratory and an association organized for the sole purpose of commercial testing. I feel that if the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS should get into the commercial testing field (and they probably must eventually get into it) we probably must organize a department to take care of that and doubtless build a laboratory for that sole purpose; and that until such time arrives the present laboratory should devote their attention to the preparation of standards under which the commercial laboratory can operate.

PRESIDENT LEWIS: The regulations before this body are presented by the Committee on Research with the idea of bringing together all the resolutions that the Society has passed, correlate them, eliminate conflicts, so that we may definitely know the limits to which we can go in testing under Society mandate. Nothing new is being started but just an attempt to get the rules and regulations under which the research should be conducted into a businesslike document. If there is no further discussion the motion before the house will be voted upon.

The vote was unanimous in favor of the motion.

L. A. Harding then introduced the newly elected Chairman of the Committee on Research, Prof. F. B. Rowley.

Professor Rowley said: "I think you will all agree that I should feel apprehensive in this new job, especially in view of the discussion that has just taken place in regard to commercial testing. As I look at these resolutions, however, they have not opened up the way for a more extensive program of commercial testing, but have merely put on some additional regulations under which such testing may be done.

It seems to me that the research which the Society has sponsored in the past has been responsible, to a large degree, for the position which this Society now holds with other corresponding professional societies. I do not believe that any group would want the Society to spend time with routine commercial testing. When such testing is necessary in order to establish fundamental principles or laws, the Laboratory should be free to make the tests. On the other hand, if it is developing into a pure routine type of testing, that kind of testing which is going to be used for advertising purposes or which is merely competing with established commercial laboratories, certainly we want to keep out of it. As far as I am concerned with the Committee on Research, I shall be glad to have the advice of those who have been responsible in the past for guiding the research work and who have built it up to its present standards.

In the past, I understand that a large amount of work has fallen to the Chairman of the Research Committee. In fact, when I accepted this job, Mr. Harding told me I might as well leave all other work and spend my time this year with the Committee on Research. The Council has, however, come to the rescue and appointed an Executive Secretary.

I appreciate the honor the Committee has conferred upon me and I assure you I realize the magnitude of the job, and I hope to have the full cooperation of everybody in keeping the work of the Committee on Research on a high plane.

Report of Certified Public Accountant on Research Fund

RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

29 WEST 39TH STREET, NEW YORK CITY

Gentlemen:

As requested I audited the books of account and records of the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Pittsburgh, Pa., for the year ended December 31, 1929, and submit herewith my report.

The recorded Cash Receipts for the year previously stated were traced into the banks, also the cancelled checks and duly approved disbursement vouchers were checked against the cash book.

A verification of the Cash on Deposit was obtained by communicating directly with the depositories and reconcilement of the amounts reported to the balances reflected by the books of account.

Respectfully submitted,

FRANK G. TUSA,
Certified Public Accountant.

CASH RECEIPTS AND DISBURSEMENTS

RESEARCH LABORATORY OF THE AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS—PITTSBURGH, PA.

FOR THE YEAR ENDED DECEMBER 31, 1929

Balance—December 31, 1928, per Former Report			$ 4,910.03
Receipts			
From the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS	$23,168.67		
Other Contributions	10,257.87	$33,426.54	
Interest on Bank Balances		110.05	33,536.59
			38,446.62
Disbursements			
Salaries—per Schedule		23,141.32	
Traveling—F. C. Houghten	1,663.95		
Traveling—Staff	200.26		
Traveling—Executive Committee	226.50	2,150.71	
Laboratory Equipment and Supplies		1,836.74	
University Cooperation		5,812.30	
Office Expenditures			
Telephone and Telegraph	126.61		
Postage	152.28		
Professional Services	100.00		
Expressage	79.24		
Photostats	74.66		
Stenography Service	57.00		
Meeting Expense	50.30		
Furniture and Fixtures	49.27		
Printing and Stationery	27.25		
Miscellaneous	25.47	742.08	33,683.15
Balance—December 31, 1929			$ 4,763.47

DISPOSITION OF FUNDS

At New York, N. Y.			
Bankers Trust Company			
Cash on Deposit		$ 1,680.54	
Securities—3M General American Tank Car Corp. Bonds (Market Value $2,910.00)		3,045.61	$ 4,726.15
At Pittsburgh, Pa.			
Oakland Savings and Trust Company			
Cash on Deposit		20.23	
Petty Cash Fund		17.09	37.32
			$ 4,763.47

The report of the Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers, was read by L. A. Harding, chairman, and is printed in full on p. 35.

Report of Guide Publication Committee

The best report that the Guide Committee can make is that THE GUIDE will be waiting for you when you get home. The GUIDE 1930 is a new book. Almost every page has been changed in one way or another. There are several new chapters and many new tables. Despite all of this costly mechanical service, the cost of the GUIDE 1930 has been reduced about 45 cents from the cost of the GUIDE 1929. These figures cannot be taken too seriously, of course, because we have increased production and because the cost each year involves some hangover cost of the year previous, which was on a smaller production. We are climbing the peak all the time. We are printing about 3,000 more copies for 1930 than we did for 1929.

THE GUIDE is supposed from its name to be a leader through dark places and over stony roads. In this connection, the Guide Publication Committee feels that it would fail in its duty as a committee unless it suggests to its successor committee a few of its ideas for use of THE GUIDE.

The 1929 Committee has worked harmoniously. For the things in the present edition of THE GUIDE which are commendable, credit should be given to the staff of the Society—Messrs. Hutchinson, Close, Houghten and Korman and Miss O'Neil, who have worked most zealously without regard to overtime in building THE GUIDE. For the things in the present edition of THE GUIDE which are not pleasing to you, blame should be given to the Chairman of the Publication Committee, who has been given a free hand by the Committee and who can offer no alibis. The Chairman has confidence that he will hear of his mistakes from you and can only promise to correct and amend as may be practicable.

The staff of the Society has developed to a point where in the judgment of the Committee it may be practicable to make THE GUIDE a routine production of the staff, without any so-called Guide Publication Committee, just as we now handle the JOURNAL, under the general supervision of the Publication Committee of the Council. The Guide Publication Committee suggests further that THE GUIDE in its function as a leader should promote and suggest improvement in nomenclature. The trade names of Heating and Ventilating Engineers are cumbersome and archaic. Many terms, such, for instance, as unit heater, vacuum valve, radiator, are misleading and cumbersome, and the present committee could develop improved trade nomenclature.

The Guide Publication Committee suggests finally that the term "square foot" and the term "horse power" be deleted from future editions of THE GUIDE, or at least that where "square feet" is used in pipe size and other tables that heat units be given in the parallel column, during the transition period, until gradually we can get rid of these old terms.

THE GUIDE is the product of the service of many generous people, most of them not even being members of the Society. The Publication Committee wishes to acknowledge its indebtedness to these friends.

Respectfully submitted,

S. R. LEWIS, *Chairman.*

W. T. Jones, chairman of the Finance Committee, presented the amendments to the By-Laws for consideration. These amendments were submitted to the Society through the mail and Mr. Jones explained that the two principal objectives were, to safeguard the funds of the Society, and to segregate under one heading those items which apply to the handling of the funds of the Society. The proposed amendments were read by Mr. Jones as follows:

AMENDMENTS TO THE BY-LAWS

In accordance with the provision of the By-Laws relating to Amendments, the

following changes were voted upon at the Annual Meeting 1930 held at the Benjamin Franklin Hotel, Philadelphia, Pa., January 28 to 31, 1930, and unanimously adopted.

BY-LAWS

(1) Article XIII—Amendments: to become Article XIV.

(2) Article XIII—Funds.

Section 1. All funds and moneys shall be received, deposited, invested and disbursed under the direction of the Council.

Section 2. When so directed by the Council, the Chairman of the Finance Committee shall invest such portion of any funds of the Society as determined by the Council in securities legal for the investment of funds of savings banks of the State of New York. All investments shall be approved by the Council.

Section 3. Any bequest or gift to the Society which the donor shall designate to be used, for a specific purpose shall, after acceptance by the Council, be deposited or invested in the manner provided by Sections 1 and 2, and the income or principal, as designated by the donor, used for the specific purpose designated.

Section 4. An endowment fund for research and such other purposes devoted to the art of heating and ventilating as may be determined by the Council shall be established. The interest or income from this fund shall be used each year as shall be determined by the Council. The principal shall remain intact and shall be deposited in banks or invested in securities legal for the investment of funds of savings banks of the State of New York, as determined by the Council in the manner provided by *Section 1 and 2.*

Section 5. At the beginning of each year the Finance Committee shall present to the Council a budget of estimated income and expenditures for the current year, which after approval by the Council shall govern the expenditure of Society funds for that year. Any proposed expenditure of Society funds outside of the approved budget shall be approved by the Council before the expenditure is made.

Section 6. Any money due the Society shall be collected by the Secretary, who shall enter all receipts in the books of the Society and deposit same to the Treasurer's account. The Secretary shall receive all bills against the Society, and shall present them for audit and approval to the Chairman of the Finance Committee. The approved bills shall be referred to the Treasurer which officer if he also approves the bills shall draw and sign a check payable to the account of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS for the total amount of the approved bills. The Treasurer shall present the approved bills with the check of the President of the Society for his examination. After approval of the bills by the President, that officer shall countersign the check in payment thereof, which check shall be deposited to the account of AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and known as the Secretary's account. The Secretary shall promptly draw against this account in settlement of the approved bills. The Secretary shall have the authority to pay salaries, traveling expenses and petty cash in accordance with the budget. In case of disability or absence of the Treasurer, the Chairman of the Finance Committee is authorized to sign checks. He shall give bond in the same manner as provided for the Treasurer.

Section 7. After December 31st, and before the Annual Meeting of the Society in January, the accounts of the Society shall be audited by a certified public accountant, selected by the Council at its last meeting in the calendar year. The auditor's report shall be presented at the Annual Meeting of the Society by the Chairman of the Finance Committee.

Section 8. The funds of the Research Laboratory shall be handled separately from the funds of the Society, in accordance with the By-Laws of the Society and the regulations which govern the Research Laboratory.

(3) Article VII, Section 4, reads as follows:

The Treasurer shall have the custody of all the funds of the Society, and shall deposit them to the credit of the Society in such bank or depository as the council may designate, and he shall disburse the same as provided in Article VIII, *Section 3.* He shall enter regularly in books of the Society to be kept by him for the purpose, full and accurate account of all moneys received and disbursed for the Society. He shall at all reasonable times exhibit his books and accounts to any member of the Council, and shall perform all duties incident to the office of the Treasurer, subject to the control of the Council. He shall give a bond in a penal sum and with a surety or sureties approved by the Council, for the faithful performance of his

duties as Treasurer. If a surety company bond is furnished the premiums therefore shall be paid by the Society.

To be amended to read as follows:

The Treasurer shall have the custody of all the funds of the Society, as provided in Article XIII. He shall at all reasonable times exhibit his books and accounts to any member of the Council, and shall perform all duties incident to the office of the Treasurer, subject to the control of the Council. He shall give a bond in a penal sum and with a surety or sureties approved by the council, for the faithful performance of his duties as Treasurer. If a surety company bond is furnished the premiums therefore shall be paid by the Society.

(4) Repeal *Section 7,* Article VI; *Section 5,* Article VII; *Section 3,* Article VIII; which are covered in the proposed Article XIII.

Article VI, *Section 7,* which reads as follows:

The Council shall designate the bank or depository in which the funds of the Society shall be deposited and shall by appropriate resolutions designate the purpose for which the funds may be withdrawn and authorize such withdrawal. In case of disability or absence of the Treasurer the Chairman of the Finance Committee is authorized to sign checks. He shall give bond in the same manner as provided for the Treasurer.

Article VII, *Section 5,* which reads as follows:

The accounts of the Treasurer and the books of the Society shall be audited annually by a certified public accountant selected by the council at least thirty (30) days before the close of the fiscal year.

Article VIII, *Section 3,* which reads as follows:

The Finance Committee shall prepare an annual budget of expenditures for the Society and shall pass upon and approve in writing all expenditures authorized by the Council. No expenditures are to be made by the Secretary, except for salaries, traveling expenses and petty cash, unless authorized by the Chairman of the Finance Committee, on forms provided for that purpose.

Amend Article VIII, *Section 1,* as follows:

Eliminate item "e" Research Committee.

Amend Article IX, *Section 1,* by substituting for the words: Article VIII, *Section 3,* the words: Article XIII, *Section 6.*

The installation of the newly elected officers of the Society, L. A. Harding, *President;* W. H. Carrier, *First Vice-President,* and F. B. Rowley, *Second Vice-President,* and the four new *members of Council,* was conducted with the assistance of Past Presidents J. I. Lyle and H. M. Hart.

The President next called for unfinished business and resolutions presented by J. F. Hale were adopted.

Resolved that the Society express its hearty congratulations to those who made possible the remarkable success of the Heating and Ventilating Exposition.

Resolved that the Society express its appreciation for the splendid cooperation they have received from the representatives of the press.

Resolved that we extend to the management of the Benjamin Franklin Hotel, our thanks and appreciation for the excellent service rendered and the pleasant accommodations provided.

Resolved that it is the sense of this meeting that the general arrangement committee involving program, transportation, banquet and the entertainment of our ladies has overlooked nothing, and we wish to extend to them our appreciation for the graceful manner in which our interests and happiness have been anticipated.

At the conclusion of the Meeting attention was called to the recent death of John A. Quinn, President of the *National Association of Master Plumbers,* and a resolution expressing the sincere sympathy of the Society to the members of his family was presented by H. G. Black, and adopted.

The meeting adjourned.

PROGRAM 36TH ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

BENJAMIN FRANKLIN HOTEL, PHILADELPHIA, PA.

Monday, January 27, 1930

10:30 A.M.	Reception of Guests and Registration Time set aside for visitors to accept invitation of the heating and ventilating manufacturers of Philadelphia and vicinity to inspect manufacturing processes and plants.
2:30 P.M.	Council Meeting
4:00 P.M.	Ladies' Tea
6:00 P.M.	Ladies' Dinner
8:00 P.M.	Heating and Ventilating Exposition, Commercial Museum
9:00 P.M.	Theatre Party for Ladies
11:30 P.M.	Buffet Supper

Tuesday, January 28, 1930

BENJAMIN FRANKLIN HOTEL

9:00 A.M.	Reception and Registration
9:30 A.M.	Greeting by John Cassell, Honorary Chairman Response by President Thornton Lewis Introduction of R. C. Bolsinger, General Chairman Report of Tellers Power from Process and Space Heating Steam, by L. A. Harding Report of Council Report of Committee on Increase in Membership, C. W. Farrar, Chairman.
12:00 NOON	Ladies' Luncheon
12:30 P.M.	Luncheon for Society Officers and Authors of Papers
1:30 P.M.	Motorbus to Valley Forge for Ladies

Tuesday, January 28, 1930

COMMERCIAL MUSEUM

2:00 P.M.	Report of Finance Committee, W. T. Jones, Chairman Pressure Difference Across Windows in Relation to Wind Velocity, by J. E. Emswiler and W. C. Randall Air Infiltration Through Various Types of Brick Wall Construction, by G. L. Larson, D. W. Nelson and C. Braatz
6:30 P.M.	Dinner Meeting Committee on Research—Benjamin Franklin Hotel
8:00 P.M.	Heating and Ventilating Exposition for Members and Ladies, Commercial Museum

Wednesday, January 29, 1930

BENJAMIN FRANKLIN HOTEL

9:30 A.M.	Report of the Committee on Research Report of Research Director Effects of Air Velocities on Surface Coefficients, by F. B. Rowley, A. B. Algren and J. L. Blackshaw Absorption of Solar Radiation in Its Relation to the Temperature, Color, Angle and Other Characteristics of the Absorbing Surface, by F. C. Houghten and Carl Gutberlet Preventing Condensation on Interior Building Surfaces, by Paul D. Close
12:30 P.M.	Ladies' Luncheon at the Benjamin Franklin Hotel
2:00 P.M.	Ladies' Matinee

Wednesday, January 29, 1930

COMMERCIAL MUSEUM

2:00 P.M. Standard Code for Testing and Rating Steam Unit Heaters, D. E. French, Chairman

Suggested Method of Testing Unit Heaters Suitable for Field Use, by L. S. O'Bannon

The Measurement of the Flow of Air Through Registers and Grilles, by L. E. Davies

4:00 P.M. Ladies' Tea, Benjamin Franklin Hotel

6:30 P.M. Past Presidents' Dinner
Dinner for Wives of Past Presidents

8:00 P.M. Heating and Ventilating Show—Commercal Museum

Thursday, January 30, 1930

BENJAMIN FRANKLIN HOTEL

9:30 P.M. Report of Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers, L. A. Harding, Chairman

Rating of Heating Boilers by Their Physical Characteristics, by C. E. Bronson

Report of Committee on Garage Ventilation, E. K. Campbell, Chairman

Report of Committee on Natural Ventilation

Airation Studies of Garages, by W. C. Randall and L. W. Leonhard

11:30 A.M. Breakfast-Bridge for Ladies, Benjamin Franklin Hotel

Thursday, January 30, 1930

COMMERCIAL MUSEUM

2:00 P.M. Report of Guide Publication Committee, S. R. Lewis, Chairman

Pipe Sizes and Orifice for Small Gravity Circulation Hot Water Heating Systems, by E. G. Smith

Panel Warming, by L. J. Fowler

Development of a Method for Heat Regulation, by F. I. Raymond and R. D. Lambert

4:00 P.M. Ladies' Tea, Benjamin Franklin Hotel

7:00 P.M. Annual Banquet and Dance, Benjamin Franklin Hotel

Friday, January 31, 1930

BENJAMIN FRANKLIN HOTEL

9:30 A.M. Amendments to By-Laws

Installation of Officers

Friction Losses and Observed Static Pressures in a Domestic Fan Furnace Heating System, by A. C. Willard and A. P. Kratz.

Air Conditioning the Halls of Congress, by L. L. Lewis and A. E. Stacey

Tests of Disc and Propellor Fans, by A. I. Brown

Resolutions

Shopping Tour for Ladies

1:30 P.M. Council Meeting

COMMITTEE ON ARRANGEMENTS

R. C. Bolsinger, *General Chairman;* John D. Cassell, *Honorary Chairman*

Finance Committee: Lee Nusbaum, *Chairman*

Ladies' Entertainment Committee: M. C. Gillett, *Chairman*

Banquet Committee: A. J. Nesbitt, *Chairman*

Publicity and Transportation Committee: F. D. Mensing, *Chairman*

Hostesses: Mrs. Thornton Lewis, Mrs. M. C. Gillett, Mrs. W. G. Culbert, Mrs. M. F. Blankin, Mrs. R. C. Bolsinger, Mrs. C. G. Binder, Miss Edgar, Mrs. A. H. McDade, Miss Theodora Jellett, Mrs. William Miller, Mrs. A. J. Nesbitt, Mrs. Lee Nusbaum, Mrs. H. J. Walther, Mrs. Warren Webster, Jr.

No. 848

REPORT OF CONTINUING COMMITTEE ON CODES FOR TESTING AND RATING STEAM HEATING SOLID FUEL BOILERS

THIS committee was charged by the Society to review and report on:

1. A.S.H.V.E. Code for Testing Steam Heating Boilers (adopted January, 1929).

2. A.S.H.V.E. Code for the Rating of Heating Boilers Burning Solid Fuel (adopted January, 1929).

The committee suggested several minor changes in the test code at the Summer Meeting 1929 which were adopted by the Society and designated A.S.H.V.E. Codes 1 and 2. Standard and Short Form Heat Balance Codes for Testing Low Pressure Steam Heating Solid Fuel Boilers. It also presented an additional test code known as the A.S.H.V.E. Performance Test Code of Steam Heating Solid Fuel Boilers, which was likewise adopted by the Society. The committee presented a suggested revision in the rating code at the Summer Meeting with the further suggestion that no action be taken at that time.

In the following report the committee has endeavored to review briefly and discuss several suggested methods of rating boilers, also the warm air furnace rating code in the following order:

1. A.S.H.V.E. Code for Rating Heating Boilers Burning Solid Fuel.

2. Recommended revision of the A.S.H.V.E. Rating Code.

3. A Rating Code as adopted by the Steel Heating Boiler Institute, December 10, 1929.

4. A rating code for warm air furnaces for gravity circulation, the first edition of which was approved by the Society in 1923.

Comments on the Committee Concerning the Subject of Boiler Rating

Object of Rating Codes

The committee believes that the principal objects of any rating code are:

(*a*) To furnish the purchaser with sufficient information in reference to the apparatus rated that he may make an intelligent selection to suit his particular needs and requirements.

(*b*) It is further apparent that the information conveyed by the rating should be given in some uniform manner in order that an intelligent price comparison may be made between similar apparatus as manufactured by different concerns.

Assuming that this general conception of a rating code is correct, it would appear that it is difficult, if not altogether impossible, to entirely divorce the subject of selection from rating. A dual responsibility exists between the purchaser and manufacturer if any installation is to be successful from the standpoint of operation. The manufacturer is responsible for furnishing correct, reliable and sufficient information regarding the apparatus and the purchaser is evidently responsible to the extent of utilizing the apparatus in a manner which is consistent with the information as furnished.

If the purchaser is provided with a uniform method for determining the loads under which the apparatus is to be employed, as for example the A.S.H.V.E. Code of Minimum Requirements for Heating and Ventilation of Buildings (Section V) which specifies the manner by which the design load and maximum load are to be determined, then it is apparent that it is incumbent upon the manufacturer to furnish at least this much guaranteed information regarding his apparatus, if the purchaser is to fulfill his part of the assumed obligation for a successful installation which it must be assumed is the object sought by all concerned in the matter.

It is apparent that the adoption of any rating code which does not permit the provisions of the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings being carried out would require a revision of this code as now written.

The following extract from the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings is frequently referred to in this report:

SECTION V

MINIMUM CAPACITY AND INSTALLATION REQUIREMENTS FOR LOW PRESSURE STEAM AND HOT WATER HEATING BOILERS

employing solid fuels stated in Btu per hour shall be taken as the sum of the following items:

(*A*) The estimated heat emission in Btu per hour of the connected radiation, direct, indirect or both, to be installed as determined by computation from data given in Sections II and III for normal operation.

(*B*) The estimated maximum heat in Btu per hour required to supply water heaters or other apparatus to be connected to the boiler or boilers.

(*C*) The estimated heat loss in Btu per hour of the piping connecting radiation and other apparatus with the boiler or boilers (see Table 1).

(*D*) The estimated increase in the normal load in Btu per hour due to starting with cold piping and radiation. This increase is to be based on the sum of items (*A*), (*B*) and (*C*) and shall be assumed not less than the following:

TABLE 1—PERCENTAGE INCREASE TO BE ADDED TO NORMAL LOAD

Sum of Items (*A*), (*B*) and (*C*) in Btu per Hour	Equiv. Steam Radiation Sq Ft (240 Btu per Sq Ft)	Percentage Increase to Be Added
Up to 100,000	Up to 417	65
100,000 to 200,000	417 to 834	60
200,000 to 600,000	834 to 2502	55
600,000 to 1,200,000	2502 to 5004	50
1,200,000 to 1,800,000	5004 to 7506	45
Above 1,800,000	Above 7506	40

(1) *Estimated Boiler Load:* For the purpose of this Code the estimated connected load to the boiler or boilers.

(2) ***Boiler Capacity to Be Installed:*** **The boiler or boilers to be installed shall be guaranteed by the manufacturer of the boiler to be capable of supplying, at the boiler outlets, the total Btu per hour as computed by the method outlined in the preceding paragraph and under the following conditions of operation each of which is to be stated in the specifications covering the installation for which the boiler or boilers are intended.**

$$\text{The Heating-up Factor} = 1 + \frac{\text{Table Percentage}}{100}$$

It appears to this committee that a rating code for any type of apparatus must of necessity include the following items:

1. The output.
2. Specified conditions of operation for the output stated.
3. The limits placed on certain specified conditions.

It is also obvious that one must reproduce the conditions in practice under which the apparatus was originally rated if comparative results are to be realized or attained. It is necessary to standardize limits for some of the specified conditions, otherwise a rating code could not fulfill its function. Some of the limits to conditions are naturally now set by custom and usage or by ordinances or laws, designed to protect the health and safety of the community. For example, custom and usage have decreed that the conditions relative to steam pressure for rating purposes, shall be 2-lb gage at the boiler for heating boilers, and 2 per cent priming, whereas, overall efficiency, draft tension, temperature of flue gas and rate of combustion are conditions over which it is difficult at least to assign exact or definite limits in the present state of this art.

Placing a minimum value limit on boiler efficiency covering the average load period of the heating season, would appear to be in line with a program of fuel conservation and would probably receive a welcome by prospective owners of heating boilers. Not the least item, however, in boiler economy is the manner in which the boiler is actually operated by the owner and over which the manufacturer has no actual control. The manufacturer can and does produce boilers which give relatively high efficiencies when properly operated but it is difficult to set a minimum efficiency limit that would produce the actual results apparently desirable.

This Committee is fully cognizant of the fact that no rating code can be devised that will make proper boiler selection for estimated loads automatic or will entirely remove the possibility of making errors in selection. The same degree of intelligence must apparently be employed in selecting a boiler from manufacturers' data that the purchaser displayed in the more intricate calculation of the heat losses for the correct selection of the amount of radiation to be installed.

DEFINITIONS

There are a number of terms employed in this art, the meaning of which is either vague, not clear or by no means standardized. In order to avoid any misunderstanding of the interpretation of the various terms employed in this discussion, the following definitions have been included:

Purchaser: Construed to mean the person responsible for the selection of the boiler.

Equivalent Direct Radiation: The heat emission of 240 Btu per hour per square foot of manufacturers' rated surface of direct steam radiation and 150 Btu per hour per square foot of manufacturers' rated surface of direct hot water radiation.

One-Number Boiler Rating: A single rating stated for each boiler listed in a manufacturer's catalog.

Multi-Number Boiler Rating: Two or more ratings stated for each boiler listed in a manufacturer's catalog.

Dimensional Boiler Rating: A one-number rating based on some physical dimension of the boiler, as for example the grate surface, boiler heating surface, or both.

Heating Boiler Output: As defined by the A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers (adopted June, 1929. See A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 332.)

Grate Area: As defined by the A. S. H. V. E. Code for Testing Low Pressure Boilers Burning Solid Fuel. (1929 edition. See A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 322).

Boiler Heating Surfaces: The sum of the areas of all surfaces in the boiler which are exposed to the products of combustion on one side and water on the other side measured in square feet.

Boiler Efficiency: The over-all efficiency of grate and boiler as defined by the A. S. H. V. E. Performance Test Code for Steam Heating Solid Fuel Boilers (June, 1929 edition. See A. S. H. V. E. TRANSACTIONS, Vol. 35, 1929, p. 332).

Priming: The amount of free moisture carried by the dry saturated vapor delivered by the boiler outlets stated as a percentage of the sum of the weights of dry saturated steam plus the free moisture delivered at boiler outlets.

Estimated Design Load: The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined and is the sum of the heat emission of the radiation to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system. (A. S. H. V. E. Code of Minimum Requirements for the Heating and Ventilation of Buildings).

Estimated Maximum Load, Peak Load, Starting-Up Load: These terms are considered synonymous and are construed to mean the load, stated either in Btu per hour or equivalent direct radiation determined by the purchaser to be the greatest estimated output that the boiler will be called upon to carry in operation. The maximum load is a function of the time assumed to raise the temperature of cold piping and radiation to normal operating temperature. The committee recommends the use of the term *estimated maximum load* in this connection. (See Section V, A. S. H. V. E. Code of Minimum Requirements for the Heating and Ventilation of Buildings.)

Net Load: This term as now employed in heating literature has various meanings, *viz.*, installed direct radiation, design load and maximum load. The committee recommends that this term be dropped from heating literature as serving no especially useful purpose.

Estimated Average Load: The estimated average load stated in Btu per hour of equivalent direct radiation for the heating season and based on the average outside temperature during the heating season for the locality in question.

Heating-Up Factor: The factor by which the estimated design load is multiplied in determining the estimated maximum load. This is equal to $1 + \frac{\textit{Percentage added}}{100}$. (See Section V, A. S. H. V. E. Code of Minimum Requirements for Heating and Ventilation of Buildings.)

Also see *Time Analysis in Starting Heat Apparatus*, by Ralph C. Taggart, A. S. H. V. E. TRANSACTIONS, Vol. 19, 1913, for a complete mathematical treatment of this subject. This paper gives heating-up factors or per cent overload based on the design load for various lengths of time required to warm up or start the heating apparatus. The author calls attention in this paper to the fact *that Boiler Performance Curves are desirable.*

The heating-up factor is dependent upon the time permitted or assumed for completely heating all the piping and radiation to operating temperature. The shorter the time the greater the boiler output required. If approximately two hours are allowed for this operation with cast iron radiation, the heating-up factor approaches a value of 1 and the boiler is therefore subject to no overload from this cause.

Heating-up factors as determined by Section V of the Code of Minimum Requirements for the Heating and Ventilation of Buildings are, it is believed, considered reasonably satisfactory by heating engineers.

The following approximate analysis of the heating-up load for a steam system although not strictly accurate, will serve to illustrate the fact that normally the maximum load on any heating boiler is determined by the time that is allowed or assumed to raise the temperature of cold radiation to normal operating temperature.

The following assumptions are made:

Weight of radiation and piping: 8 lb per square foot of installed equivalent steam radiation. Specific heat of iron: 0.12. Initial temperature of iron: 40 F. Final temperature of iron: 215 F. Temperature of air surrounding radiation: 40 F and assumed constant during the heating-up period. Unit heat emission for radiating surface: 1.7 Btu per hour per square foot per degree difference in temperature between radiating surface and the air. Heat emission of radiation: 240 Btu per hour per square foot in normal operation. Time allowed for the heating-up period: 45 minutes or ¾ hour.

To heat the iron requires: $8 \times 0.12 \ (215-40) = 168$ Btu per square foot of radiation. The heat emission of the radiation during the heating-up period is approximately:

$$\frac{3}{4} \times 1.7 \left(\frac{215 + 40}{2} - 40 \right) = 111.6 \text{ Btu per square foot}$$

The total heat to be supplied by the boiler in ¾ hour is therefore:

$168 + 111.6 = 280$ Btu, or at the rate of
$280 \times 4/3 = 373$ Btu per hour.

The boiler for the assumed conditions evidently must be capable of delivering $\frac{373}{240}$ or 1.55 (heating-up factor) times as much heat as is required for normal operation.

Attention, Firing Period, Fuel Available in Hours: The hours required to burn one available fuel charge. Fuel available is defined in the A. S. H. V. E. Performance Boiler Test Code for Steam Heating Solid Fuel Boilers. (See 1929 edition A. S. H. V. E. TRANSACTIONS Vol. 35 1929 p 332). The committee recommends the discontinuance of the use of the terms *attention* and *firing period*.

(1) A.S.H.V.E. CODE FOR RATING HEATING BOILERS BURNING SOLID FUEL (ADOPTED JANUARY, 1929)

The rating of a boiler under this code requires that a series of tests be conducted under the rules of the A.S.H.V.E. Code for Testing Steam Heating Boilers to determine the actual output of the boiler covering a number of rates of combustion with a fuel having a calorific value of 12,500 Btu per pound and corresponding recorded flue gas temperatures. These two items are plotted against output on a chart which is part of the Code.

The intersection of the combustion rate curve and flue gas temperature curve with corresponding lines printed on the chart each correspond to some output. The A.S.H.V.E. Rating output is the lower of the two outputs determined in this manner, plus 30 per cent of their difference providing the priming does not exceed two per cent moisture and that the CO_2 in the flue gases is not less than 12 per cent by volume.

The Code states: Rating outputs conforming to this Code shall be known as A.S.H.V.E. Rating.

This Code fixes the *maximum output* that may be designated as the A.S.H.V.E. Rating. The output allowed by this Code may require a higher draft or other operating characteristic than the manufacturer would desire when listing the boiler for average use; this Code allows that a lower rating output (assumed maximum) may be chosen and listed as the A.S.H.V.E. Rating.

This scheme evidently provides a ***one number maximum output rating*** as determined by providing certain limits on the operating conditions. The following paragraph appears under the heading

Purpose:

This Code is not intended to supplant the more correct engineering practice of determining the available output of boilers for specified operating conditions by a study of analysis of the complete data given by *performance charts* nor does it preclude the assigning of other rating output values to meet purchasing specifications or a specified set of operating conditions. The rating outputs as determined by this Code are intended for average conditions and for the use of purchasers not competent or desirous of making comparisons and selections from performance charts.

The intent of the last sentence in the preceding paragraph is not entirely clear. The purchaser provided with *only* the maximum output of the boiler would necessarily have to be sufficiently competent to select and apply a safe heating-up factor to his estimated design load to arrive at the maximum load in order to select the correct boiler rated in this manner, or the maximum output rating as given by the manufacturer would have to be divided by the purchaser's assumed heating-up factor to arrive at the output rating corresponding to the estimated design load. In either case it must be assumed that the purchaser is sufficiently competent to apply the provisions of Section V of the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings.

The committee interprets the term *performance charts* to include performance tabular data from which charts may be constructed. The committee infers that a boiler rated in this manner is not to be connected with a greater maximum load than is determined by the provisions of Section V, A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings. A more detailed discussion of this Code appears later in this report.

(2) SUGGESTED REVISION OF A.S.H.V.E. RATING CODE

This revision is intended to provide a Code which requires the manufacturer, if he employs the A.S.H.V.E. Rating, to supply sufficient tabular output data for the construction of performance charts, which it is believed is in accord with the trend of modern engineering practice. The suggested revision requires a minimum of five outputs having a range from maximum output listed to at least 35 per cent of maximum output for the minimum output listed.

The only limit placed on the operating conditions for the outputs listed is that priming shall not exceed 2 per cent. Under each output listed the proposed revision requires numerical values for each of the following items:

1. Fuel available.
2. Combustion rate.
3. Efficiency.
4. Draft tension.
5. Chimney dimensions.
6. Flue gas temperature.

This scheme provides the purchaser with complete information as to the performance of the boiler under conditions as specified. It is recommended that the manufacturer print in bold face type or otherwise designate one of the output ratings listed for each boiler which will correspond with the provisions of Section V of the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings. This feature also automatically provides a one-number design-load output rating.

Furthermore, the purchaser with a multi-rating system is afforded the opportunity not provided by a single number rating, of selecting or comparing boilers to suit his individual idea and requirements as to economy, firing period, draft, or height of chimney available, etc. The committee believes that the purchaser of any apparatus is entitled to full information in regard to its operating char-

acteristics. A more detailed discussion of the proposed code appears later in this report.

DISCUSSION OF A.S.H.V.E. CODE FOR RATING HEATING BOILERS BURNING SOLID FUEL (ADOPTED JANUARY, 1929)

The committee in applying this Code to the rating of those boilers for which performance data were made available by boiler manufacturers found in many cases that either the combustion rate performance curve from the test data, or the flue gas temperature performance curve, or both, did not intersect the corresponding combustion and flue gas temperature limitation lines as contained in the Code.

Extrapolation of the performance curves beyond the test data in practically all cases examined gave a rating larger than the manufacturers' catalog rating. This condition is natural as the manufacturer's rating is perhaps more often intended to correspond to the design load to which the boiler is to be connected and *not* the maximum load to be carried during the heating-up period.

Figures* 1, 2, 3 and 4 shows the application of the A.S.H.V.E. Code for Rating Heating Boilers Burning Solid Fuel to several boiler performance charts. Figs. 6 and *6a* show the application of the A.S.H.V.E. Code for Rating Heating Boilers Burning Solid Fuel to four cast-iron sectional type boilers of the same make. It will be observed that the maximum output as determined by this Code parallels with and is fairly close to that as reported by the manufacturers for maximum output. In both cases the rate of combustion *decreases* as the grate area and output *increases.*

The application of the A.S.H.V.E. Rating Code to six cast-iron magazine type boilers of the same make is shown by Figs. 7 and *7a.* It will be observed that the maximum output rating as determined by the Code requires a *decreasing* rate of combustion as the grate area and output increases. The rate of combustion for the manufacturer's design output rating and maximum output rating, however, increases with the increase in output and grate area.

Referring to Table 2 and Fig. 5, it will be observed that when the present rating Code is applied to determine the *maximum output,* and the Code of Minimum Requirements for the Heating and Ventilation of Buildings is employed to determine the output corresponding to the design load, in many cases this latter output compares favorably with the manufacturers' catalog rating for boilers below 3,000 sq ft rating. Above this rating, however, the comparison is not so obvious. The committee believes it is impractical at the present time at least, to assign limits to a combination of combustion rate and flue gas temperature, without considering the CO_2 limit, which would be reasonably equitable to the many and varied designs of heating boilers now available or in contemplation. The committee can find no precedent in which a condition affecting the efficiency of the apparatus is limited. Limits placed on flue gas temperature and CO_2 would undoubtedly have an effect on efficiency. *It is believed by this Committee that it is the prerogative of the manufacturer to decide on the maximum output rating he desires to publish for any boiler he manufactures.*

*Figs. 1 to 7a, inclusive, and Table*2 appear in Continuing Committee Report published and mailed to members in advance of the meeting for discussion.

The corresponding *design load* to which the boiler may be connected is determined by dividing the maximum output rating by the heating-up factors as given by the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings employing the same chimney. It is not likely that a manufacturer would willfully specify or require the installation of a chimney height beyond customary practice for the sole purpose of obtaining a high rate of combustion and therefore increased maximum output rating. See Section VII, A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings for minimum height of chimneys.

RECOMMENDED REVISION OF THE JANUARY, 1929, A.S.H.V.E. CODE FOR RATING STEAM HEATING SOLID FUEL HAND-FIRED BOILERS

(1) Purpose

The purpose of this Code is to standardize the method to be employed and followed by any person, partnership, firm, corporation or association, who may desire to make use of or employ for any purpose whatsoever the statement: "The rating of the boilers herein listed are in accordance with the provisions of the A.S.H.V.E. Code (year) for Rating Steam Heating Boilers Burning Solid Fuel."

(2) Rating Designation

It is understood that all ratings stated are guaranteed boiler outputs by the manufacturer for the corresponding boiler designation as were determined and defined by the provisions of the A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Hand-Fired Boilers (1929) and as governed by the conditions as set forth under paragraph (4) and accompanying the ratings.

The output for each boiler shall be stated in thousands Btu per hour and also in square feet of equivalent direct radiation. It shall be optional to state, in addition to the two methods indicated, the output in pounds of steam per hour.

(3) Range of Outputs for Each Boiler Listed

There shall be stated a minimum number of five boiler outputs for each boiler listed. The outputs shall have a range from maximum output to approximately 30 per cent of the maximum output and the intermediate outputs given are to be approximately equally spaced between the minimum and maximum outputs.

The manufacturer shall print in bold face type or otherwise designate the output rating of each boiler corresponding to the design load to which the boiler is intended to be connected as determined by the provisions of Section V of the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings.

(4) List of Conditions, Statements of Limiting Conditions and Manufacturers' Guarantee

There shall be stated under each output listed the numerical values for each of the following five items:

1. Fuel available in hours.
2. Combustion rate, pound per hour per square foot of grate surface.
3. Over-all efficiency, per cent.
4. Average draft tension, inches of water.
5. Interior dimensions of chimney and height.
6. Average flue-gas temperature, degrees Fahrenheit.

The following statements shall be included with the rating tables: "The priming for any output listed does not exceed two (2) per cent."

For Anthracite Fuel

"The ratings are based on a steam pressure of 2-lb gage at the boiler and anthracite coal stove size, having a calorific value of 12,500 Btu per pound on a moisture-free basis."

For Bituminous Fuel

"The ratings listed are based on a steam pressure of 2-lb gage at the boiler and bituminous coal 3 in. by 2 in. size, having a calorific value of 13,000 Btu per pound sulphur content not exceeding 2 per cent and volatile content of not less than 30 per cent on a moisture-free basis."

For Coke Fuel

"The ratings listed are based on a steam pressure of 2-lb gage at the boiler and by-product or gas coke of commercial size best suited to the boiler."

"The inside dimensions and height of chimneys listed should be satisfactory when properly constructed and having no other opening except for the purpose of serving the boiler and when free from the effect of adverse air currents. Allowance should be made for any other chimney openings, elbows in the smoke flue or breeching and for extra long smoke flue or breeching."

(5) TABLE OF DIMENSIONS

A comprehensive table of dimensions of the boilers listed shall be included in the same bulletin or catalog with the ratings. This table shall include the number and pipe size of steam and return connections and location, smoke flue dimensions and height above floor line, grate area and height of boiler-water line and such other dimensions as may be required for properly indicating the boiler to scale on a set of complete heating installation plans.

(6) DEFINITIONS

Purchaser:

Construed to mean the person responsible for the selection of the boiler.

Manufacturer:

The individual, firm or corporation who manufactures the boilers for which corresponding ratings are listed.

Boiler Output:

As defined by the proposed A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Estimated Maximum Load:

Construed to mean the load stated in Btu per hour or equivalent direct radiation that has been estimated by the purchaser to be the greatest or maximum load that the boiler will be called upon to carry.

Estimated Design Load:

The load, stated in Btu per hour or equivalent direct radiation, as estimated by the purchaser for the conditions of inside and outside temperature for which the amount of installed radiation was determined and is the sum of the heat emission of the radia-

SUGGESTED ARRANGEMENT OF PRESENTING OUTPUT DATA

BOILER DESIGNATION	Output Btu per Hr Output Eq. Steam Rad. Sq Ft Output Eq. Water Rad. Sq Ft	1255000 5220 8367	864,000 3,600 5,760	700000 2920 4667	563000 2355 3754	466000 1940 3106
S-O-4 Grate area = 11.94 sq ft Available fuel holding capacity=627 lb.	Fuel available, hours	4	6.4	8	10	12
	Rate of combustion, pounds per hour	13.12	8.5	6.56	5.25	4.37
	Efficiency, per cent	64.0	69.6	71.5	72.0	71.5
	Average draft, inches of water	0.14	0.12	0.06	0.03	0.02
	Avarage flue-gas temperature, degrees fahrenheit	540	502	460	420	380
	Inside dimension chimney, inches	16x20	16x16	16x16	12x16	12x12
	Min. height of chimney, feet	55	50	45	45	45

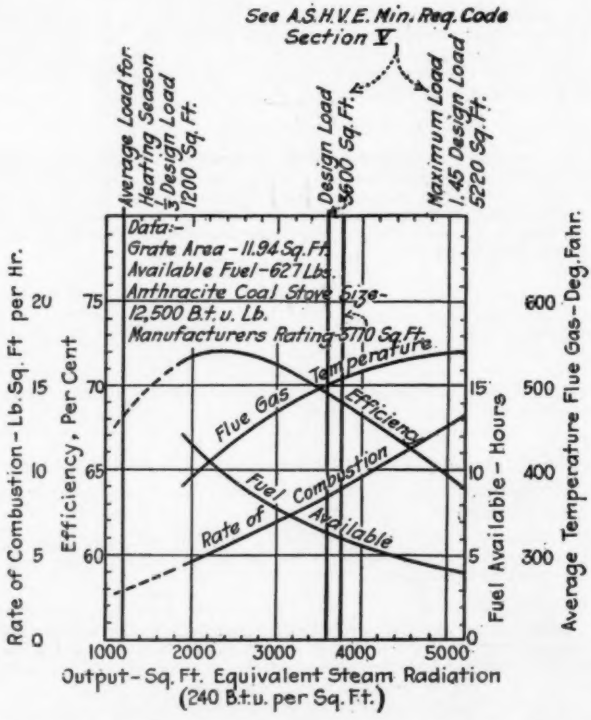
(Statements as specified by the proposed revision of the code to accompany the rating tables)

It is recommended that the manufacturer call attention to the fact that if the boiler is to be able to satisfactorily develop the maximum listed output the corresponding size chimney must be employed.

tion to be actually installed plus the allowance for the heat loss of the connecting piping plus the heat requirement for any apparatus requiring heat connected with the system. (A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings.)

Equivalent Direct Radiation:

Construed to mean the heat emission of 240 Btu per sq ft of manufacturers' rated



EXAMPLE OF BOILER PERFORMANCE CURVES PLOTTED FROM BOILER PERFORMANCE DATA GIVEN IN THE TABLE

surface of direct steam radiation and 150 Btu per hour per sq ft of manufacturers' rated surface of direct hot water radiation.

Grate Area:

As defined by the A.S.H.V.E. Code for Testing Low Pressure Steam Heating Solid Fuel Boilers.

Fuel Available in Hours:

Construed to mean the hours required to burn one available fuel charge. The available fuel is defined by the proposed A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Over-All Efficiency:

As defined by the proposed A.S.H.V.E. Performance Test Code for Steam Heating Solid Fuel Boilers.

Priming:

The amount of free moisture carried by the dry saturated steam vapor delivered by the boiler stated as a percentage of the total weight of the sum of the dry saturated steam plus the free moisture delivered.

Signed L. A. HARDING, *Chairman,*
R. V. FROST
F. C. HOUGHTEN
CONTINUING COMMITTEE ON CODES FOR TESTING AND RATING STEAM HEATING SOLID FUEL BOILERS.

JOINT DISCUSSION

Report of Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers

and

Rating of Heating Boilers by Their Physical Characteristics

By C. E. BRONSON

(*See page 225*)

W. H. SEVERNS (WRITTEN): Mr. Bronson stated that the plotted data shown in Figs. 1 and 2 are those pertinent to more than 800 boilers. These questions immediately arise:

(1) Why was the straight line of Fig. 1 drawn through the upper plotted points to fix the capacity of a steel boiler at 14 sq ft of standard radiation per square foot of boiler heating surface?

(2) Why was the curve of Fig. 2, fixing the capacities in square feet of standard radiation for various grate areas, drawn above the average of the plotted points for the larger boilers?

Both curves as drawn represent ratings considerably greater than those that the majority of steel boiler manufacturers have been willing to ascribe to their various products.

The rating of 14 to 17 sq ft of standard steam radiation per square foot of heating surface cannot always be justified by the numerical computations shown for power boilers, *i.e.,* where 10 sq ft of heating surface may be allowed for a boiler output of 33,524 Btu per hour, or the equivalent of 140 sq ft of standard radiation per 10 sq ft of heating surface. Large power boilers may easily equal or exceed this output, while smaller boilers may never under conditions equal the foregoing performance.

The capacity obtainable from a boiler under fixed operating conditions is dependent upon the circulation of the water within the boiler, the amount of heating surface available for the absorption of the heat liberated by the burning fuel, the location of the heating surfaces, the amount of grate surface

upon which the combustion of the fuel may take place. Two boilers may not develop the same capacity when operated under identical conditions as to fuel, combustion rate, etc., although each has the same amounts of heating and grate surfaces. If the circulation of the water is hampered in one boiler and not in the other, the boiler with the restricted circulation will develop less capacity, likewise the same is true if the equal amounts of heating surfaces are so placed that one boiler is handicapped by the location of its heating surfaces.

Steel heating boilers are built in sizes ranging from those having about 1.8 sq ft of grate area to those having more than 60 sq ft of grate area. Such equipment ranges from toy size to that of a full-fledged power boiler. Obviously then, the combustion spaces and the heating surfaces are likely to be deficient in amount in the smaller units. In small boilers the ratio of the heating surface to the grate surface may vary from 10 or 15 to 1, while in large units the ratio may be 50 to 1.

Boiler performance is vitally affected by the boiler design and its proportions. All other conditions being fixed, a boiler cannot be satisfactorily given a rating based upon its physical dimensions alone, if mention is not made of the kind and calorific value of the fuel to be used, the combustion rate, the grate area, and the operating efficiency.

The Rating Code for Steel Heating Boilers ignores nearly all of the foregoing essential conditions. The purchaser of a code rated steel boiler has no idea of what may be necessary to make the boiler deliver its rated capacity. Furthermore, if a boiler so rated does develop the rated capacity at the steam outlets, the purchaser has nothing to indicate the operating efficiency of the unit. Will the boiler be an efficient or inefficient absorber of the heat liberated from the burning fuel? Will the listed capacity be obtained by wasting the fuel?

Knowledge of the capacities and the corresponding efficiencies at the different combustion rates possible with different available drafts is essential in order that the purchaser may properly select a boiler to operate under the conditions of his plant. Such information can be secured only by actual rating tests, and not by empirical equations based upon one or two physical dimensions of the boiler. The Steel Boiler Code is an attempt to rate all steel boilers, irrespective of their design or efficiency of operation, on the basis of 14 to 17 sq ft of radiation capacity per square feet of heating surface. Two empirical equations are provided for the calculation of the required grate area necessary for the capacity based on the heating surface. These equations seem unnecessarily complicated. The equations may be transposed without mathematical alteration to give the rating for a given amount of grate area. The equations with the terms transposed are:

Catalog rating in square feet $=$ (25.5 $\times$ grate area squared) $+$ 200.

Catalog rating in square feet $=$ (16.8 $\times$ grate area squared) $+$ 1500.

Either of the equations increases the rated capacity very rapidly as the grate area is made larger (see Fig 2 of paper). Properly placed and sufficient boiler heating surfaces are more important than grate area is, if efficient boiler operation is to be secured. Small ratios of heating surface to grate surface are not desirable. It would seem that the general tendency will be to code rate boilers on their grate areas, without regard to the boiler design or the location or the amounts of their heating surfaces.

We have plotted, at the University of Illinois, a number of curves based on the catalog data of several modern cast-iron and steel boilers. For the cast-iron boilers there have been no published data relative to the amounts of the boiler heating surfaces. Some of the cast-iron boilers have a very high rated capacity per square foot of grate area, which means that the rating must be secured by undue forcing of the boilers, high temperatures of the flue gases as they escape, lowered efficiency, and wasteful use of the fuel.

W. H. CARRIER: All I have to say is largely in support of the report of the last and especially the present Committee. I want to compliment Mr. Bronson on his very able presentation. He has gone about it in a scientific manner and I think he has put the arguments for the dimensional rating as clearly and forcibly as they can be put.

From a manufacturer's standpoint, where the forms and types of equipment are thoroughly conventional, such dimensional rating tends to maintain such standards, such proportions of design as are then in use. As, for example, take our rating of the automotive engine. For a long time this rating was considered a fair representation of performance. What is it today? In this case it did not have the effect of standardizing because people were not buying automobiles on their dimensional ratings, but on their performance.

Now in a boiler industry, the steel boilers have a great similarity to each other; they are quite similar in type and they will at least lend themselves, we must admit, more easily to dimensional rating than any other. The question is, are we to have one rating for steel boilers and another for cast iron boilers, or will some new type come up? I can see it is for the interest of established industry to maintain the present line of manufacture. Is it for the interest of the public, or is it ultimately for the interest of the manufacturer? That is a question that is open for debate. I do not know. I would think that, inasmuch as Mr. Bronson's very able engineering discussion of the relationships of present dimensions to ratings shows that performance, which is a thing that they require, is quite definitely related to dimensions, that at least a rating based on performance rather than dimensions would be more fundamental than the ratings which are based on dimensions only or derived from performance. They must be.

He mentioned the 10-sq ft per horse power rating of the old tubular boilers, power tube boilers and water tube boilers of some kind. Well, that was a sort of a Chinese law that I think did more to retard the development of boilers than probably any other thing. Manufacturers had a hard time to convince users that where they were getting as a general rule 40 per cent more out of a given surface by improved construction, their boiler would do the work. They were penalizing advanced design simply as a matter of prejudice. Very fortunately that 10 sq ft rating has long since gone by the board completely. The progressive engineer today selects his boiler on known performance, depending somewhat on the type, and in every type of surface differs somewhat and the conditions of operation are primarily important.

It is true that you probably have a standardized conventional design at present in which relations of performance to dimensional data are quite exact, but what will we have five years from now, or even two years from now? Maybe we will have the same and maybe we will have some change. We do not want to throttle this Society from advancement. Therefore, when we

base on performance, I believe we are on sound grounds and we are not penalizing any boiler whose relation of performance to dimensional ratios are fixed. More convenient to go on dimensional relations, but no penalization.

It will just point out one fact. A radiating surface is many times more effective than an indirect surface. Suppose we change the relations of these, which is now fairly standardized, which is not for cast iron, nor is the relation between the two definite. It is practically impossible for any formula Mr. Bronson makes to evaluate the effectiveness of the direct surface as compared with the other complicated formula, and yet it is absolutely vital on performance. We are basing our results on performance. Why that thing is wiped off the slate and we are really simplifying the whole situation when we base on performance rather than entirely on dimensional rating.

It is true we must have certain fire-box areas, certain grate areas and depths for firing capacities. That is a practical limitation except where we have stokers or use other fuel, but so far as the surface itself is concerned, I have just had occasion to review, for example, some effects of indirect surface, change in proportions, shape of tubes, dimensions of tubes, and with the same velocities through the tubes, low velocities, it happened to be in this case. There was almost twice the heat transfer per degree difference on this indirect surface, due to different form, different dimensions, spacing, than there was in the other case, which shows a great variation may be expected by changes in design which are basic and where the same square feet of surface is maintained.

Now, the effect of that surface depends on dimension. A small tube, for example, will give you more results per given surface than a large tube. An elongated tube will give you more than a round tube. If you were going to go into passes, you will get more result from two passes than you will from one. If you change your velocities you get a still further effect. When you get above a certain critical point; you have lost it over your surface.

So this is a very complex thing when you go into the design of boilers that Mr. Bronson is doing, to know where you are going to get at. If this Society is going to make a code on dimensions, it is merely good as far as present design is concerned and does not in any way apply to future design. As long as it is admitted that a performance rating will answer, I think it is a mistake to go to dimensional rating, which answers only particular cases and may be changed. That is the principal argument that I have for basing it on the committee's report rather than purely on the dimensional rating.

R. V. FROST: In the discussion of these codes, it is very necessary to keep in mind the trade relationship that these codes have upon the manufacturers and upon the trade. The Society cannot very well oppose the steel boiler manufacturers when they are as a unit ready to adopt a dimensional code, if that serves their trade conditions. In the same way the Society cannot oppose the cast-iron boiler manufacturers upon a code that meets their peculiar conditions.

The performance method type of code has received the approval of practically all cast-iron boiler manufacturers. I think there are only one or two of these manufacturers who are not taking a definite stand in favor of that code, and in the same way they are just as strongly opposed to the one number method of rating, for the reason that the one number rating is bound up

so intimately with the pricing and the selling of the boiler. We can adopt a one number rating code but it need not be used, because it cannot be made compulsory. To adopt such a code means that we are going to go into the pricing of boilers and that is a phase the Society has no business to consider. We do have the right to present data or ask for data that will enable any one to properly select a boiler and that is just exactly what the performance table method of rating does.

Those manufacturers who have adopted the performance table method of rating and are using it in their catalogs now would not go back to the one number method of rating. Some use the one number method and also the performance table method, but it is to be noted that the sales organizations of those companies are resorting more and more to the performance method than to the one number method.

Within this last year at least 50 per cent of the producing capacity in the cast-iron boiler industry has taken up the performance table method and another large manufacturer is to take it up next year, which will make very close to 60 or 70 per cent of the total producing capacity using the performance table method.

For that reason the Society cannot very well do otherwise than offer them a uniform method of performance table coding, and that gives the Society a very good reason to adopt the performance table code as presented by the committee.

On the dimensional code, it has really not satisfied the steel boiler industry as a whole because the power boiler branch of the industry is working at the present time to develop some other method of rating, and they are turning more and more to a performance table method. While there has been no action taken by the power boiler branch; the idea is crystallizing among the leaders of that branch of the industry to such an extent that practically every power boiler sold, as Mr. Carrier said, is sold on performance, not upon dimension.

H. M. HART: As far as this Society is concerned, I do not see how we can do anything different than our committee have recommended. For an engineering Society to recommend and adopt a dimensional code for rating boilers would be illogical.

Insofar as the report of the committee is concerned, I am very much pleased with it. I think that it is logical, understandable, easy to interpret. I can find no fault with it, with one exception. It has no factor of safety whatever. In practical application, as I understand it, a boiler manufacturer can test his boiler up to its maximum capacity and then rate his boiler on design load at the percentage below maximum recommended in our code of minimum requirements. To think that the manufacturer is going to place his maximum output at some point below that is rather hard to imagine. He is out to sell boilers and he is going to show all that his boiler will do.

Therefore, to select boilers from these performance charts on the basis recommended by this report means that in actual practice every morning when we start up a steam system we will have to run that boiler at its maximum output, expecting to be able to duplicate on the job the ideal conditions under which the boiler was tested. A boiler selected on that basis will absolutely

have to have uniform fuel. Where is the man going to be when his fuel happens to fall below 12,500 Btu when he has selected his boiler on that basis of fuel? He is going to be out of luck, because he has no factor of safety. Somebody might leave a window open when he is starting up in the morning and he would also be out of luck. You have not even a factor of safety for that.

I made some comparisons, taking the design load as recommended by this code and as recommended by a boiler manufacturer and there is a uniform variation of from 40 to 45 per cent. In other words, the manufacturer recommends that his boiler be placed on loads that are 40 to 45 per cent below the design loads recommended by this code. How would the Boiler Output Committee of the *Heating and Piping Contractors' National Association* interpret these performance charts? I think we would take this report and place on an additional factor of 40 per cent.

I am pleased to see the recommendation that we have performance charts. Now as to the *Steel Heating Boiler Institute,* I think that a group of boiler manufacturers representing 95 per cent of the output, getting together and agreeing on a uniform basis of rating, is very commendable; I think it is fine and from a sales standpoint and from a practical standpoint I think they have done a fine job and I congratulate them. As far as this Society is concerned, why, of course, we could not lower ourselves to the point of recognizing anything so unscientific as dimensional rating.

J. D. CASSELL: In order to bring the matter properly before the Society I move you that we adopt the code as presented by the committee. (The motion was seconded.)

MR. NEWCOMB: I would just like to bring this thought out, that the committee's report and Mr. Bronson's paper and all of the discussions that have preceded have shown a sharp difference of opinion between the designers of heating boilers, cast-iron boilers and steel heating boilers. The average size steel heating boiler installed is approximately 4000 ft. Under that size undoubtedly cast-iron boilers are installed predominantly. Steel boilers go into larger size jobs. It would seem to me that these discussions have shown a lack of common meeting ground between the designers of cast-iron and steel boilers. Therefore, the code recommended by this committee, which we understand from Mr. Frost has been accepted by a number of the manufacturers of cast-iron boilers, should not apply to the manufacturers of steel heating boilers. The Society should not have this code apply to those boilers; since the manufacturers of steel heating boilers have got together and pretty nearly accepted the dimensional code it would seem to me that the report of Mr. Harding should be limited perhaps to the smaller boilers and to the cast-iron boilers and either the code not apply to the dimensional code be accepted by the Society for the larger and the steel boilers.

MR. RUSSELL: I want to correct an expression that has been made this morning in that the steel heating boiler as it is manufactured and sold today is not a power boiler. It is a heating boiler. The *Steel Heating Boiler Institute* is comprised of 95 per cent of the steel boiler manufacturers that are in the country. This code that we have adopted as manufacturers is being used by over 50 per cent of the manufacturers today and they have their catalogs based on these ratings. Of the other 50 per cent probably most of

them have their catalogs in the process of printing. Our objection to the Society's Code is that primarily we want to conserve the ratings or maintain rather the conservative ratings of steel heating boilers that have been used with such success over a period of years.

MR. HART: I would like to see a factor of safety put in there when it comes to a selection of boilers. I think it is dangerous to put it out without it. I do not like to make an amendment to the motion, but I would like to have the committee consider that before this thing is adopted.

MR. SEELIG: I am not a boiler manufacturer, or I have not any connection with them. My business is selecting boilers for my clients. I would like very much to see this Society go on record on dimensional code. We have had a little experience in the last two or three years which has been rather disappointing to me in taking the boilers rated on performance. If there is any boiler on the market today that is rated on performance that we can safely go and select a boiler on their figures, I would like to find out where it is. For that reason I would hate to see the steel boilers thrown over into the same disorganized position that the cast-iron boilers are. We have still got to adopt some pretty large factors of safety ourselves or else we will get into trouble.

L. A. HARDING: I would like to comment on one or two of the statements that have been made. First, Mr. Carrier brought out the fact clearly that there is a vast difference between the heat absorption value of shine and indirect surface. There is no question about that.

Who knows, next week or next year what the trend of boiler design is going to be? I doubt if any one here could answer that question. I heartily concur with the speaker who brought out the fact that we cannot afford to adopt a code that is not going to adequately protect the Society in the future.

I think Mr. Hart made the statement that the method of determining the design load recommended by this code is not a safe procedure. This particular code does not recommend any set method for determining the design load. We recommend, that the manufacturer publish in his performance tables the design load in accordance with A.S.H.V.E. minimum requirements code. The code does not prevent any one who has occasion to specify a size of boiler for a heating system to use any higher factor or added percentage he may desire. If you have a performance table in front of you and desire to compare boilers on a basis of efficiency or flue gas temperature, etc., you will be able to satisfactorily make a comparison except on a basis of boiler heating surface. If you desire to compare a boiler on the basis of heating surface, write to the manufacturer. They would be glad I am sure to furnish you with the necessary data for such comparison. I do not suppose that a cast iron boiler manufacturer would object to furnishing the direct heating surface of the boiler.

One gentleman made the remark that he would like to know where performance curves could be obtained on boilers that he could use. There are several manufacturers now employing performance data derived from tests for rating purposes.

Mr. Obert made the suggestion that the committee endeavor to correlate the two proposed boiler rating codes, that is the A.S.H.V.E. Committee recommended revision for a boiler rating code and the *S.H.B.I.* method of dimen-

sional rating. This might be possible if all makes of steel boilers were exactly alike in design and the same ratio of direct to indirect heating surface would always remain the same.

I would rather question the advisability of tying the Society to a method of rating that we cannot safely back up. It is rather doubtful to suppose that one could expect the same efficiency for the same output from two boilers having the same total heating surface but with different ratios of direct to indirect surface.

F. B. ROWLEY: I do not care to discuss the boiler rating code, as I believe its contents have been clearly brought out by previous speakers. It is evident that the Society is interested in getting a code by which boilers may be rated on a scientific basis, and, at the same time, one which will not obstruct future progress or development. The question of a boiler code has been before this Society for a long time. It has been revised, referred back to committees, and I believe the consideration given it has not been exceeded by any other code which has been before the Society. The present Committee and the Council have given the code as it now stands very careful consideration and have approved it.

There is a motion before the house that the code be adopted. I would like to move that this motion be referred to the members for a letter ballot vote. Since the code has received so much discussion and is of interest to the membership at large, a greater majority of which are not present, I believe this would be the most satisfactory method of settling it.

The motion was duly seconded and carried and referred to the Secretary of the Society for action.

No. 849

POWER FROM PROCESS AND SPACE HEATING STEAM

By L. A. Harding,[1] Buffalo, N. Y.

MEMBER

THE necessity for the utilization of recoverable wastes and what are termed by-products in practically every industry becomes increasingly apparent each year and is primarily due to competition and plant location.

The utilization of a particular recoverable waste in one plant may not, owing to the relatively advantageous location of the plant, be either practical or profitable, whereas, a competitor, on the other hand, finds that it is not only profitable but an economic necessity. The mere fact that a waste is recoverable is no criterion that its recovery is either desirable or a profitable venture. A new process is rarely, if ever, so perfect that it is not susceptible to economic improvement by recovering some form of waste whether it be in the form of a material product or a reduction in the fuel or electric power bill. It so happens in one industry, at least, that the price of the principal product is largely determined by the market value of the by-product from the process.

A given amount of money spent for improvement of the process, the utilization of more up-to-date machinery, etc., may show a better return on the investment than the same amount spent on the equipment required for a waste recovery. It is not generally a simple matter to estimate with any great degree of accuracy the equivalent money value from the recovery of a by-product, the amount and value of the product saved or recovered being subject for various reasons to considerable fluctuation.

Electric power rates in various parts of the country show a great variation. Low fuel cost and a high power rate are naturally a combination that would appear to be the most desirable condition for power recovery from process steam. This combination however rarely ever exists. Investigation covering cost estimates for the recovery of solids, liquids, vapors, the heat equivalent of fuel and power are all receiving the attention of progressive manufacturers. This paper, as the title indicates, deals only with the specific problem involved in the investigation of the generation of electric power from process steam.

Wherever steam is employed for process work the boiler pressure is determined by that part of the process requiring the highest temperature, and generally the bulk of the steam is employed at a considerably lower temperature

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and corresponding pressure. The heat available per pound of available process steam, or so-called *heat drop,* for the generation of power is the difference between the heat content i_1 of the steam at the generating pressure (initial condition) and the heat content i_2 of the steam corresponding to the pressure at which

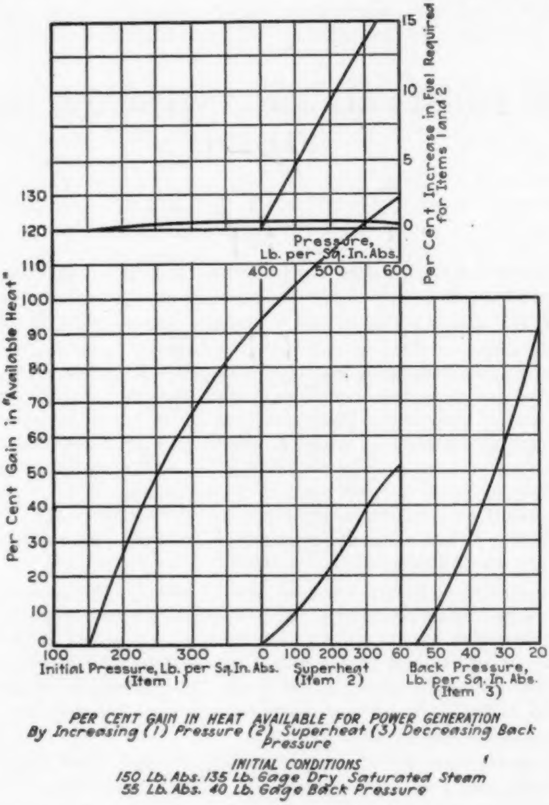


FIG. 1. PER CENT GAIN IN HEAT AVAILABLE FOR POWER GENERATION

it is employed in the process (final condition), assuming adiabatic expansion (Rankine Cycle). This difference multiplied by the pounds of steam so employed per hour gives the total heat available for power generation by means of reciprocating engines or steam turbines.

The more or less fixed amount of process steam available at the existing process pressure is perhaps more often found to be inadequate to supply all of the power requirements and some means should be adopted to secure as near a balance between the heat available and the heat equivalent of the power re-

quirement as it is possible or practical to obtain with a minimum amount of expense for extra fuel. The heat available $(i_1—i_2)$ per pound of steam may be increased by (1) increasing the initial pressure (2) superheating and (3) decreasing the process pressure (back pressure).

The comparative effect on the basis of per cent gain in the heat available per pound of steam of the foregoing items as based on an existing condition of 150 lb per square inch absolute pressure (135 lb gage) boiler pressure as required for a small part of the process dry saturated steam and 55 lb per square inch absolute (40 lb gage) pressure for the bulk of the process steam assumed available for the generation of power, is shown by Fig. 1. The *per cent gain* is evidently the difference between the available heat per pound of steam for the existing condition and the proposed condition divided by the available heat per pound for the existing operating condition.

Suppose, for example, there was found, on comparing the available process steam load and the power load curves and assuming a certain prime mover to be employed, a deficiency of approximately 15 per cent of the steam available for power generation. Fig. 1 indicates that this deficiency can be met in any one of three ways, viz:

1. Increasing the boiler pressure to 175 lb per square inch absolute or an increase of 25 lb per square inch.

2. Superheating the steam 150 deg.

3. Reducing the process (back) pressure to 47 lb per square inch absolute or a reduction in the process pressure of only 8 lb per square inch.

Increasing the boiler pressure would in all probability require the installation of new boilers, an expensive procedure. Superheaters, if the plant is equipped with water tube boilers, could be installed at nominal expense but would require approximately 7 per cent more fuel for superheating. The reduction in back pressure, if at all possible, would obviously be the natural procedure, as the slight difference in temperature resulting from the reduction in pressure would probably have comparatively little effect on the process. If, however, the deficiency in process steam available for power generation amounted to say 40 per cent, then it is apparent that the boiler pressure must be increased to 225 lb per square inch absolute or increased a smaller amount and some superheat employed, or use 310 deg superheat with no increase in pressure, or reduce the process pressure to 36 lb per square inch absolute (21 lb gage), a reduction of 17 lb per square inch. In this case the installation of new boilers designed for the higher pressure and equipped with superheaters would probably be the outcome.

Frequently higher process pressures than necessary are carried and some experimenting to this end may return exceedingly good dividends. The natural procedure in the investigation is to start in the reverse order, and it is frequently found that at least two or possibly all of these gains must be employed to secure the desired result. There remains the possibility of increasing the amount of power that may be generated without additional expenditure for fuel by substituting an electric motor for the small steam engines or steam pumps employed in the plant.

The possibility of effecting boiler plant economies, tending to offset any additional fuel as may be required when items (1) and (2) are considered.

should not be overlooked. A study of the existing plant requirements covering both the steam and electric power load curves may result in suggested methods to reduce the peak loads resulting in savings that likewise cost nothing to secure except the time required for the survey and analysis.

The investigation of any specified case generally involves a number of assumptions for the purpose of determining the most economic initial pressure and superheat for the required back pressure as determined by the process. The installation of superheaters for the existing boilers, of additional boiler capacity to be operated at a higher pressure in conjunction with straight non-condensing turbines, mixed pressure, extraction or bleeder type turbines, heat exchangers, steam regenerators or accumulators, and condensing operation utilizing hot well

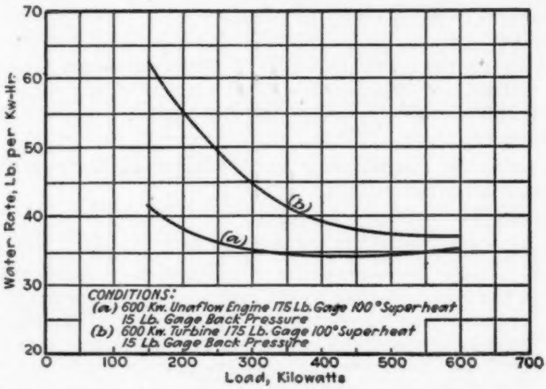


FIG. 2. WATER RATE CURVES FOR NON-CONDENSING OPERATION

water as a source of hot water supply, etc., are some of the many items which are frequently considered in this connection.

A knowledge of the water rates of prime movers for various conditions of operation is evidently essential to anticipate the amount of power that may be generated from the available process steam. The water rates of high grade reciprocating engines operating non-condensing are somewhat less than the steam turbine of equivalent capacity when operating with the same initial and back pressure. The steam turbine, when operating condensing, however, shows as good an economy as the reciprocating engine.

The turbine offers the advantages that steam may be extracted at various pressures in the expansion, occupies less floor space and, combined with the electric generator, is usually a less expensive combination. There are cases, however, when the most economical machine is essential to give a profitable balance between the heat and power requirements. A comparison of the guaranteed water rate curves for a 600 kw combination of a uniflow engine and generator (*a*) and a steam turbine and generator (*b*), both operating with an initial pressure of 175 lb per square inch gage at the throttle, 100 deg superheat and 15 lb per square inch gage back pressure is shown by Fig. 2.

The rated capacity of generating units adaptable for use in the great majority

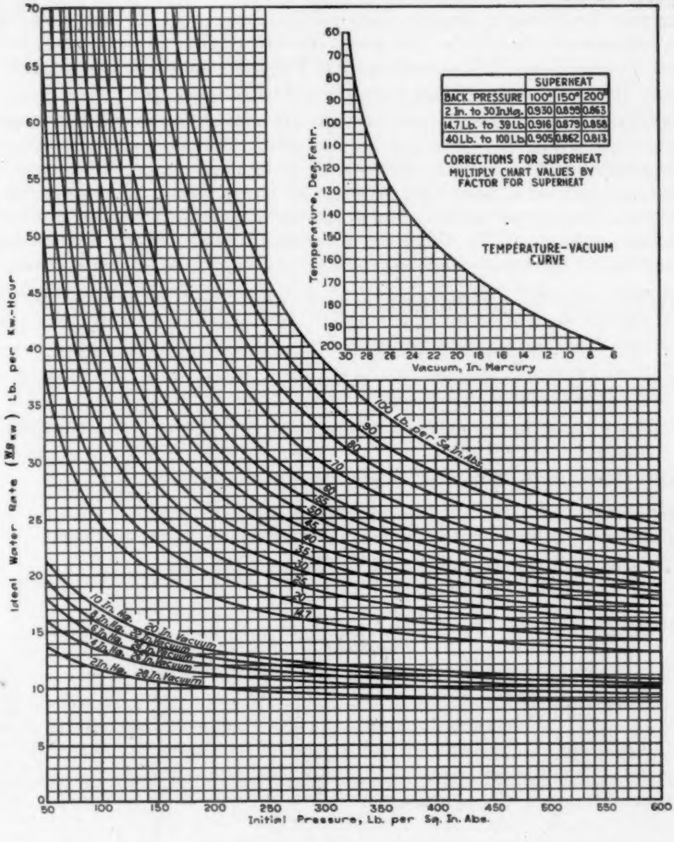


FIG. 3. IDEAL WATER RATE PER KILOWATT HOUR WITH DRY SATURATED STEAM FOR VARIOUS INITIAL PRESSURES AND BACK PRESSURES

of manufacturing plants ranges between 200 and 1,500 kw. The data in this paper refer particularly to the use of steam turbines within these capacities. Steam turbines especially designed to meet exacting requirements naturally are somewhat more expensive and more economical in the use of steam over a limited range of operating loads than a standard commercial design. Standard commercial designs, however, will usually show as good an average economy for fluctuating load condition, and are ordinarily employed when process steam is utilized for power generation.

A steam turbine connected with a generator is rated on the basis of equivalent kilowatt output at full load. The rating of the generator is given in kilovolt-

amperes (kva) with a stated power factor (usually 80 per cent). A 937.5 kva generator, for example, would require a turbine rating of 937.5x0.80 or 750 kw. The guaranteed water rates for steam turbine generating sets are universally stated in pounds per kilowatt output not including the small amount of electric energy required for excitation with alternating current generators.

The theoretical or ideal water rate of a steam engine or turbine employing steam expansively is determined on the assumption that the expansion takes place adiabatically (constant entropy). The Rankine cycle has for many years been employed as a basis for purposes of comparison. It is assumed, with this cycle, that complete expansion of the steam takes place between the initial and back pressure. The following formula is employed for determining the theoretical or ideal water rate for either steam engines or steam turbines:

i_1=heat content in Btu per pound of steam at an initial pressure on turbine side of throttle of p_1 lb per square inch absolute. (The usual pressure drop assumed through throttle valve is 15 lb per square inch.)

i_2=heat content in Btu per pound of steam at the back (terminal) pressure p_2 lb per square inch absolute.

h=heat available for conversion into work in Btu per pound of steam.
$=i_1-i_2$.

778=mechanical equivalent of heat (foot-pounds per Btu).

1.34=ratio of kilowatt to equivalent electrical horse power.

$W\,R_{hp}$=Ideal water rate—pound per brake horse power per hour.

$W\,R_{kw}$=Ideal water rate—pound per kilowatt per hour.
$=1.34\ W\,R_{hp}$

$$W\,R_{hp}=\frac{33000\times 60}{778\ h}=\frac{2546}{h}\qquad(1)$$

$\emptyset_t$=Brake Potential Efficiency ratio of turbine (see table 5).

$$=\text{ratio of}\ \frac{\text{ideal water rate}}{\text{actual water rate}}$$

$\emptyset_g$=Efficiency of generator (see Fig. 4).

$E\,W\,R_{hp}$=Expected water rate of turbine-pound per brake horsepower per hour.

$$=\frac{W\,R_{hp}}{\emptyset_t}=\frac{2546}{\emptyset_t\times h}\qquad(2)$$

$E\,W\,R_{kw}$=Expected water rate-pound per kilowatt hour.

$$=\frac{E\,W\,R_{hp}}{\emptyset_g}=\frac{2546\times 1.34}{\emptyset_t\times\emptyset_g\times h}=\frac{3412}{\emptyset_t\times\emptyset_g\times h}\qquad(3)$$

The brake potential efficiency ratio takes into account all losses for the turbine and is a combination of the internal efficiency $\emptyset_i$ and the mechanical efficiency $\emptyset_m$ or $\emptyset_t=\emptyset_i\times\emptyset_m$. The average mechanical efficiency for the turbines from 300 to 2000 kw capacity may be assumed as 94 per cent to 96 per cent, so that $\emptyset_i=\frac{\emptyset_t}{0.94}$ (approx.). The internal efficiency is employed for determining the terminal point for the expansion line of the turbine when drawing this line on a Mollier diagram.

The small and medium size steam turbines employed for the generation of power for industrial plants are of the following types:

NON-CONDENSING OPERATION

The back pressure employed varies approximately from 10 to 50 lb per square inch gage, although there are installations operating with a considerably higher back pressure. Turbines for this service are of the impulse type usually with comparatively few pressure stages, two velocity stages being employed for the first pressure stage. Frequently, only one rotor is used, the steam from the nozzles being redirected through the same row of moving blades as with the re-entry type turbines or two rows of blades attached to the same rotor with a stationary set of blades located between the two rows of moving blades. This latter is termed a Curtis stage. If more than one pressure stage is employed the succeeding stages are each provided with one velocity stage, *i. e.*, one row of moving blades. Some impulse turbines, however, employ a single rotor for each velocity stage. This arrangement is generally known as a Rateau or multicellular type turbine.

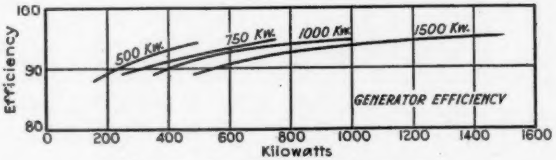


FIG. 4. GENERATOR EFFICIENCIES

CONDENSING OPERATION

Turbines for condensing operation are of the impulse type for the first pressure stage, usually with two velocity stages for the first rotor. The remaining pressure stages are each provided with a single velocity stage. The total number of pressure stages employed varies from 6 to 12 for impulse turbines (multicellular) with different manufacturers. The stages, after the first pressure stage, are either of the single velocity stage for each pressure stage (Rateau type) or a combination impulse and reaction stage. Parsons type with plurality of stages. The chart (Fig. 3) gives the values of $W\ R_{kw}$ covering various initial and back pressure conditions for steam initially dry and saturated. When the steam is initially superheated the values as read from the chart are to be corrected by the superheated factors given in the accompanying table. The factors are average values for the range of back pressures indicated.

Tables 1, 2, 3 and 4 give values of h and $W\ R_{kw}$ for various pressures and superheat. The values given for h were read from a Mollier diagram by the late G. A. Goodenough. The condition of the steam at various pressure stages in its passage through the turbine is conveniently determined by means of the Mollier Chart, as later shown by several examples. In order to draw the expansion line on this diagram with sufficient accuracy for the purpose at hand, approximate values for pressure stage efficiencies are generally assumed, or the method recommended by E. H. Brown and M. K. Drewry (*Trans.*

TABLE 1. HEAT AVAILABLE AND IDEAL WATER RATE LB PER KW HOUR DRY, SATURATED STEAM

Back Pressure Lb Sq In. Absolute p_2	Initial Pressure Lb Sq In. Absolute (p_1)													
	100		150		200		250		300		400		600	
	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.
29-in. Vac.	327.4	10.42	351.5	9.72	369.0	9.25	382.0	8.94	392.5	8.70	407.5	8.38	427.0	7.99
28-in. Vac.	293.0	11.65	318.7	10.70	336.5	10.13	350.0	9.75	360.5	9.47	376.5	9.07	397.0	8.60
26-in. Vac.	257.9	13.23	284.2	12.00	302.02	11.29	316.0	10.80	327.6	10.41	344.2	9.92	365.3	9.34
24-in. Vac.	236.4	14.43	262.0	13.01	281.5	12.12	295.6	11.55	306.6	11.13	324.5	10.50	346.2	9.86
22-in. Vac.	220.4	15.47	247.2	13.80	266.5	12.80	280.6	12.15	291.9	11.70	309.5	11.02	332.0	10.27
20-in. Vac.	208.1	16.40	235.3	14.50	254.5	13.40	269.0	12.68	280.5	12.16	298.5	11.43	321.0	10.63
14.7	140.0	24.37	169.3	20.15	189.5	18.00	205.3	16.60	217.9	15.65	237.5	14.36	261.4	13.05
20	120.0	28.45	149.7	22.80	170.5	20.00	186.6	18.28	199.2	17.12	218.2	15.64	244.3	13.96
25	104.4	32.70	134.7	25.31	156.0	21.88	172.0	19.83	184.9	18.45	204.0	16.72	230.0	14.83
30	91.2	37.40	121.5	28.10	143.5	23.80	159.6	21.38	172.7	19.75	192.5	17.71	219.1	15.57
35	80.4	42.50	111.7	30.55	133.0	25.66	149.4	22.83	162.4	21.00	182.7	18.66	209.3	16.30
40	70.6	48.30	101.5	33.60	123.8	27.57	140.4	24.30	153.6	22.20	174.0	19.60	201.0	16.96
45	61.6	55.40	93.0	36.70	115.2	29.61	132.1	25.85	145.4	23.45	166.0	20.55	193.8	17.60
50	54.0	63.20	85.7	39.80	108.0	31.60	124.6	27.40	138.4	24.65	159.0	21.46	186.8	18.25
55	47.0	72.60	78.7	43.40	100.7	33.86	117.8	28.98	131.6	25.93	152.5	22.37	180.3	18.91
60	.....	.....	72.2	47.20	94.5	36.10	111.6	30.60	125.4	27.20	146.5	23.30	174.8	19.50
70	.....	.....	60.7	56.20	83.3	41.00	100.6	33.90	114.6	29.80	136.0	25.10	164.6	20.73
80	.....	.....	50.0	68.20	73.0	46.70	90.6	37.70	104.5	32.66	126.4	27.00	155.3	21.96
90	.....	.....	41.0	83.25	64.0	53.32	82.0	41.60	96.4	35.40	118.0	28.90	147.6	23.10
100	.....	.....	33.2	103.0	56.3	60.66	74.1	46.05	88.4	38.60	110.2	30.95	139.8	24.40

Table 2. Heat Available and Ideal Water Rate Lb Per KW Hours—100 Deg Superheat

Back Pressure Lb Sq in. Absolute p_2	Initial Pressure Lb Sq In. Absolute (p_1)													
	100		150		200		250		300		400		600	
	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.
29-in. Vac.	346.5	9.85	373.5	9.14	393.0	8.69	407.0	8.38	418.5	8.15	437.0	7.81	460.5	7.41
28-in. Vac.	311.0	10.97	339.2	10.05	359.5	9.50	373.2	9.15	385.0	8.87	404.0	8.45	429.0	7.95
26-in. Vac.	274.5	12.43	302.7	11.27	323.0	10.56	338.5	10.08	350.5	9.74	370.0	9.23	395.6	8.63
24-in. Vac.	251.0	13.60	280.5	12.16	301.3	11.32	316.4	10.78	329.0	10.36	348.5	9.80	375.0	9.10
22-in. Vac.	235.0	14.51	264.2	12.90	285.5	11.95	300.9	11.34	313.0	10.90	333.5	10.22	360.0	9.48
20-in. Vac.	222.0	15.36	251.7	13.55	273.0	12.50	288.4	11.83	301.0	11.33	321.5	10.61	348.6	9.78
14.7	151.0	22.60	182.5	18.70	205.0	16.64	221.4	15.40	235.0	14.50	256.5	13.30	285.0	11.97
20	130.0	26.24	161.7	21.10	185.0	18.44	201.4	16.94	215.0	15.85	237.5	14.36	266.5	12.80
25	114.0	29.90	146.0	23.35	169.0	20.20	186.0	18.34	200.0	17.06	222.5	15.33	252.1	13.53
30	100.0	34.12	132.7	25.70	156.0	21.86	173.4	19.67	188.0	18.15	210.0	16.25	240.0	14.20
35	89.0	38.30	121.7	28.05	145.0	23.52	162.4	21.00	177.0	19.26	200.0	17.06	229.6	14.85
40	78.5	43.45	111.0	30.72	135.0	25.28	152.8	22.33	167.0	20.41	190.5	17.90	221.0	15.44
45	69.0	49.45	102.2	33.40	126.0	27.10	143.4	23.80	158.0	21.60	181.5	18.80	212.0	16.10
50	60.0	56.90	94.2	36.20	118.0	28.90	136.1	25.06	151.0	22.60	174.0	19.60	205.6	16.60
55	53.0	64.40	87.0	39.25	111.0	30.72	129.4	26.37	144.5	23.60	167.1	20.40	199.0	17.14
60	.....	.....	80.0	42.60	105.0	32.50	122.4	27.88	137.0	24.90	160.5	21.25	192.6	17.70
70	.....	.....	67.5	50.55	93.0	36.70	110.9	30.76	126.0	27.10	149.5	22.80	181.5	18.80
80	.....	.....	56.7	60.20	82.0	41.60	100.0	34.12	115.0	29.70	139.0	24.54	171.0	19.95
90	.....	.....	46.7	73.10	72.5	47.05	91.4	37.35	106.0	32.20	130.5	26.15	163.0	20.94
100	.....	.....	37.7	90.50	64.0	53.30	82.4	41.40	98.0	34.80	122.0	27.96	155.0	22.00

TABLE 3. HEAT AVAILABLE AND IDEAL WATER RATE LB PER KW HOUR—150 DEG SUPERHEAT

Back Pressure Lb Sq In. Absolute p_2	Initial Pressure Lb Sq In. Absolute (p_1)													
	100		150		200		250		300		400		600	
	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.
29-in. Vac.	357.0	9.56	384.5	8.87	405.0	8.43	420.0	8.13	432.5	7.89	451.0	7.57	477.0	7.15
28-in. Vac.	321.0	10.62	349.0	9.78	370.1	9.21	386.0	8.84	398.5	8.57	418.0	8.17	444.5	7.68
26-in. Vac.	283.4	12.05	312.5	10.91	334.1	10.20	350.0	9.75	363.0	9.40	383.0	8.91	410.5	8.32
24-in. Vac.	260.1	13.10	289.5	11.78	311.6	10.95	328.0	10.40	341.1	10.00	361.5	9.44	389.7	8.76
22-in. Vac.	243.4	14.01	273.1	12.50	295.0	11.56	312.0	10.94	325.5	10.48	346.4	9.85	374.5	9.12
20-in. Vac.	230.0	14.82	260.2	13.10	282.6	12.06	299.0	11.40	313.0	10.90	334.0	10.20	362.0	9.44
14.7	157.4	21.68	189.2	18.02	213.0	16.01	230.5	14.80	245.0	13.92	267.0	12.78	298.0	11.45
20	136.0	25.10	168.5	20.25	192.6	17.70	210.3	16.20	225.0	15.15	247.4	13.80	278.5	12.25
25	119.4	28.60	152.0	22.42	176.6	19.30	194.0	17.58	209.5	16.28	232.0	14.70	263.0	12.98
30	105.6	32.30	138.5	24.62	162.5	21.00	181.5	18.80	196.4	17.38	219.4	15.55	251.0	13.61
35	93.4	36.55	126.5	26.96	151.6	22.50	170.0	20.05	185.0	18.43	207.9	16.40	241.0	14.18
40	83.0	41.10	117.0	29.18	141.6	24.10	160.2	21.30	175.5	19.44	198.4	17.20	231.3	14.75
45	73.4	46.50	107.5	31.74	132.6	25.72	151.0	22.60	166.5	20.50	189.4	18.00	223.0	15.30
50	64.4	53.00	99.5	34.30	124.6	27.40	143.0	23.85	159.0	21.45	182.4	18.70	215.0	15.86
55	57.0	59.90	92.0	37.10	117.6	29.00	136.0	25.10	151.5	22.50	175.4	19.45	208.0	16.40
60	.....	.....	85.0	40.10	110.6	30.85	129.5	26.35	145.0	23.53	169.0	20.20	202.0	16.90
70	.....	.....	72.0	47.40	98.1	34.80	117.0	29.16	133.0	25.66	157.0	21.72	191.0	17.85
80	.....	.....	60.0	56.85	86.6	39.40	106.5	32.05	122.0	27.98	146.6	23.28	180.0	18.95
90	.....	.....	49.5	68.90	77.1	44.25	97.0	35.20	113.0	30.20	137.6	24.80	171.5	19.95
100	.....	.....	39.5	86.40	67.6	50.50	88.0	38.80	104.5	32.65	129.0	26.45	163.5	20.87

Table 4. Heat Available and Ideal Water Rate Lb Per KW Hour—200 Deg Superheat

Back Pressure Lb Sq In. Absolute P_2	Initial Pressure Lb Sq In. Absolute Cp_1)													
	100		150		200		250		300		400		600	
	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.	Btu	W. R.
29-in. Vac.	368.0	9.27	396.5	8.60	417.0	8.18	433.0	7.88	446.5	7.64	466.5	7.32	494.0	6.91
28-in. Vac.	331.0	10.32	361.0	9.45	382.0	8.94	398.2	8.56	412.0	8.28	432.0	7.90	460.5	7.42
26-in. Vac.	293.0	11.65	323.4	10.55	345.6	9.88	361.7	9.44	376.0	9.08	396.5	8.61	426.0	8.01
24-in. Vac.	269.0	12.68	300.0	11.37	322.4	10.58	339.0	10.05	353.5	9.66	374.9	9.11	404.0	8.45
22-in. Vac.	252.0	13.54	283.2	12.05	305.6	11.16	322.7	10.58	337.5	10.11	358.9	9.49	389.0	8.78
20-in. Vac.	238.5	14.30	270.0	12.64	292.6	11.66	310.0	11.00	325.0	10.50	346.9	9.84	377.0	9.05
14.7	164.5	20.73	198.0	17.22	222.0	15.36	240.2	14.20	255.0	13.39	279.0	12.24	310.5	11.00
20	143.0	23.85	176.4	19.33	201.0	16.96	219.2	15.55	234.5	14.55	258.4	13.20	290.5	11.75
25	126.5	26.98	159.7	21.36	184.6	18.48	202.7	16.83	218.5	15.61	243.0	14.05	275.5	12.39
30	111.0	30.70	146.0	23.37	171.0	19.95	189.7	17.96	205.5	16.60	229.9	14.85	263.0	12.98
35	99.0	34.45	133.7	25.50	159.1	21.45	178.2	19.13	193.8	17.60	218.9	15.60	252.0	13.55
40	88.0	38.80	123.7	27.60	149.5	22.80	168.2	20.30	184.0	18.55	209.0	16.32	243.0	14.05
45	78.0	43.70	114.0	29.92	139.6	24.45	159.0	21.45	175.0	19.50	199.4	17.10	234.0	14.58
50	68.0	50.20	105.0	32.50	131.5	25.95	150.5	22.67	167.0	20.44	192.0	17.78	226.2	15.10
55	59.0	57.80	98.0	34.80	124.6	27.40	143.7	23.75	160.0	21.34	184.9	18.45	219.5	15.55
60	.....	.....	90.0	37.90	117.0	29.15	136.7	25.00	153.0	22.30	178.4	19.11	213.0	16.02
70	.....	.....	76.2	44.80	103.6	32.95	124.2	27.50	140.5	24.30	165.9	20.55	201.0	16.97
80	.....	.....	63.7	53.55	92.6	36.85	113.0	30.20	130.0	26.25	155.4	21.95	190.5	17.90
90	.....	.....	52.7	64.75	81.6	41.80	102.7	33.20	120.0	28.44	145.9	23.40	181.5	18.80
100	.....	.....	42.0	81.30	71.0	48.03	93.0	36.70	110.0	31.05	137.5	24.82	173.7	19.63

A.S.M.E., 1923) is employed. The internal stage efficiency ($\varnothing_s$) as here employed is the ratio of the heat converted into work by the rotor or rotors for a single pressure stage to the heat given up ("heat drop") by adiabatic expansion of the steam between the initial and terminal pressures for the pressure stage under consideration per pound of steam.

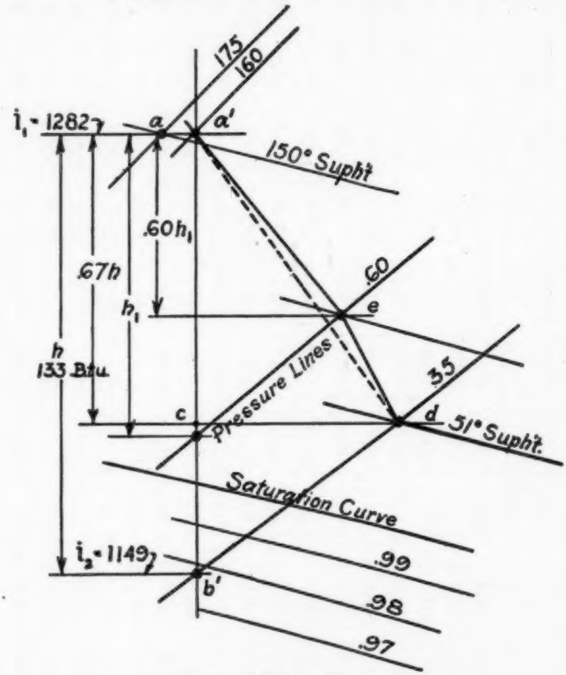


FIG. 5. MOLLIER DIAGRAM

a. State of Steam Entering Throttle Valve, 175 lb Per Sq In 150 Deg Superheat.
a'. State of Steam for 15 lb Drop in Pressure Through Valve, 160 lb Per Sq In Absolute 156 Deg Superheat.
b'. Theoretical State After Expanding from 160 lb Absolute to 35 lb Absolute Pressure.
d. Final Estimated State of Exhaust from Turbine, 35 lb Absolute 310 F or 51 Deg Superheat.

VALUES OF $\varnothing_s$

The following values of $\varnothing_s$ may be employed:

Curtis type stage (2 velocity stage to one pressure stage) usually employed in all multicellular turbines for the first stage also in conjunction with reaction turbines (Parsons type). $\varnothing_s = 0.55$ to 0.65 Pressure drop 90 to 120 lb per square inch. Rateau stages (1 velocity stage per pressure stage) employed in multicellular turbines after the first stage $\varnothing_s = 0.75$ to 0.85.

The *heat drop* per stage following the Curtis stage is generally assumed constant. There is no direct relation between the overall potential efficiency ($\varnothing_t$) of the turbine and the various stage efficiencies ($\varnothing_s$).

A straight line drawn on the Mollier diagram between the initial and final condition points as *a'-d* (Fig. 5) is frequently employed as representing the expansion line for approximate determinations of the condition of the steam at various pressure stages. The lower internal efficiency of the Curtis stage, however, generally makes it inadvisable to employ this method.

Referring to Fig. 5, it is assumed that the steam is supplied to the turbine throttle at a pressure of 175 lb per square inch absolute, as indicated on the diagram by *a*. The loss in pressure through the throttle valve is assumed at 15 lb per square inch, so that the steam enters the first pressure stage of the turbine with the same heat content at 160 lb per square inch absolute indicated as *a'* with a temperature of 518 deg (156 deg superheat). If the steam expanded adiabatically from 160 lb per square inch to an absolute pressure of 35 lb per square inch as point *b'*, *the available heat* (*h*) is the difference between the total heat for conditions *a'* and *b'* or $h=1282\ (i_1)—1149\ (i_2)=$ 133 Btu per pound of steam. $W\,R_{kw}=\dfrac{2546 \times 1.34}{133}=25.6$ (compare with value determined by means of the curves and superheat factors for 160 lb pressure and 150 deg superheat (Fig. 3).

For an assumed turbine brake potential efficiency ($\varnothing_t$) or 0.63 and generator efficiency ($\varnothing_g$) of 0.944 at full load.

$$E\ W\ R_{kw}=\frac{2546 \times 1.34}{0.63 \times 0.944 \times 133}=43 \text{ lb per kilowatt hour.}$$

Fig. 5 shows the estimated expansion line *a'-e-d* for a two pressure three velocity stage machine. It is assumed that the first pressure stage is a Curtis type stage and that $\varnothing_s$ for the first stage is 0.60. For an assumed mechanical efficiency of 0.94, $\varnothing_i=\dfrac{\varnothing_t}{0.94}$ or 0.67 for complete turbine.

THE WILLANS LINE

Referring to the diagram located at the upper left hand corner of Fig. 6, if the total pounds of steam for any two loads, as for example, the *no load* (*b*) and *full load* (*a*) are known for a throttling steam engine or a steam turbine and a straight line is drawn between these points, the total pounds of steam for any intermediate load is correctly given by the ordinate for that load. The diagram is correct when the rating is based on the brake horsepower of a turbine and the indicated horsepower for an engine. The total steam for any intermediate load may also be calculated by means of the equation:

$$S=S_1\,[x\,(1-y)+y] \qquad (4)$$

S =Total steam per hour for the desired load.

S_1=Total steam per hour for the full load.

y =Ratio of *no load* steam to *full load* brake horsepower steam.[2]

=Approximately 0.22 for non-condensing turbines.

=Approximately 0.14 for condensing turbines.

=Approximately 0.16 for low pressure condensing turbines (atmospheric pressure—82-in. vacuum).

x =fraction of full load corresponding to S.

[2] The values of y vary somewhat for various types of turbines depending on the method of governing (throttling or non-throttling) speed, pressure, number of stages, etc.

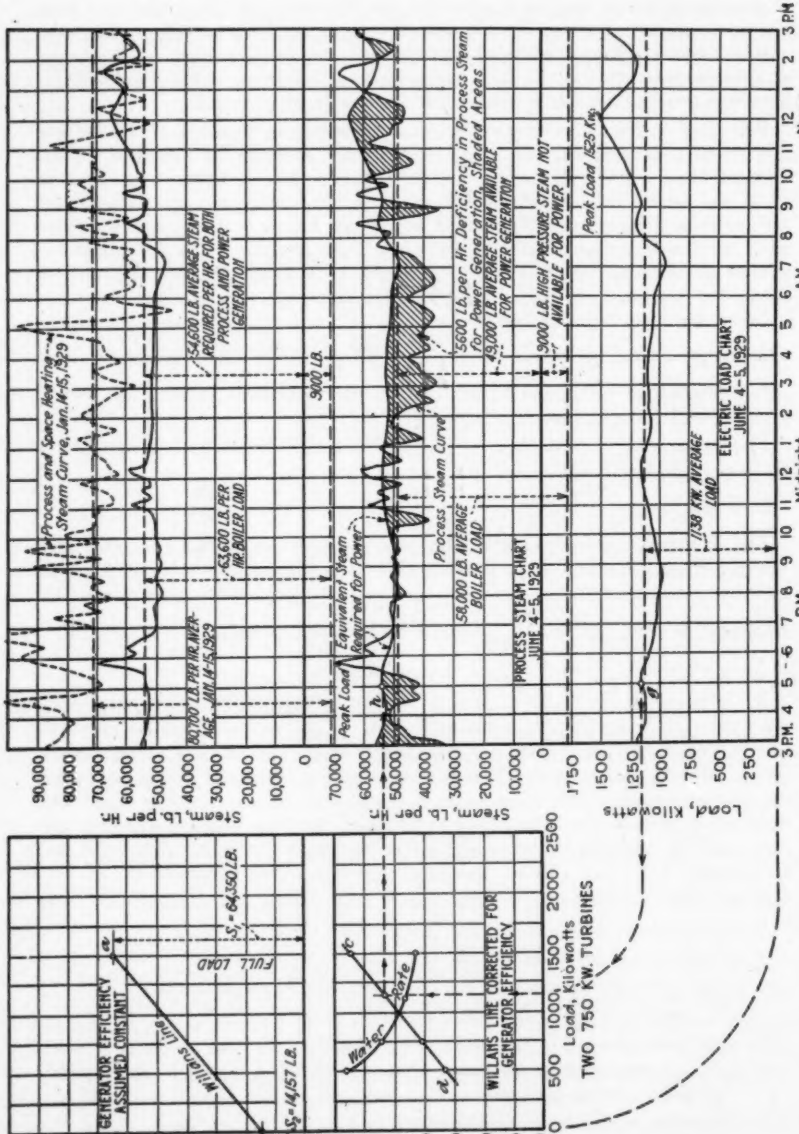


FIG. 6. STEAM AND ELECTRIC LOAD CURVES

TABLE 5. APPROXIMATE FULL LOAD EFFICIENCIES MULTI STAGE IMPULSE STEAM TURBINE—GENERATORS—GEARED TYPE

Rated Capacity at Full Load kw	Brake Potential Efficiency Ratio of Turbine $\varnothing_t$ (1)	Generator Efficiency at Full Load $\varnothing_g$ (2)	Efficiency at Full Load Combined $\varnothing_t \times \varnothing_g$ (3)
	Non-Condensing Turbines		
100	0.47	.....	.....
200	0.48	.....	.....
300	0.61	.....	.....
400	0.63	.....	.....
500	0.64	0.940	0.600
600	0.64	0.942	0.603
750	0.65	0.944	0.614
1000	0.67	0.945	0.633
1250	0.67	0.947	0.634
1500	0.67	0.950	0.637
2000	0.68	0.954	0.649
	Condensing Turbines		
500	0.68	0.940	0.639
600	0.69	0.942	0.650
750	0.70	0.944	0.661
1000	0.70	0.945	0.662
1250	0.70	0.947	0.663
1500	0.70	0.950	0.665
2000	0.70	0.954	0.668

1. Ratio $\frac{\text{Ideal Water Rate}}{\text{Actual Water Rate}}$ includes efficiency of gears.
Turbines of the re-entry type are somewhat less efficient than the values given in table for turbines 500 kw capacity and above.
2. 60 cycle—3600 rpm.

It is necessary to correct the Willans line to allow for a decreasing generator efficiency as the load is reduced below the power developed at *full load,* as will be later shown by an example.

The water rate for any load is determined by dividing the total steam per hour by the corresponding load.

The example (Fig. 6) illustrates a method employed in plotting an equivalent electric power load curve from an available process steam load curve, or an equivalent steam load curve from a power load curve. It is, of course, desirable to have the actual guaranteed water rates from the several manufacturers, whose machines are being considered, covering the proposed conditions of operation. These water rates will ordinarily be found to vary somewhat between the different types of machines proposed. The correct use of the data herein presented, however, will, it is believed, give sufficiently accurate water rates for the purpose intended. The process steam load curve and the purchased electric power load curve shown were plotted from meter readings recorded June 4 and 5, 1929, at the plant under consideration. The load curves for this date were chosen as representing a typical summer day operation when the electric power load gave a high average and maximum peak. The average steam load in this plant does not vary much from day to day during the summer months.

If it is possible to generate the electric power required by the plant from

the process steam plus an amount of steam which is less than the difference between the average winter and summer steam loads, during the summer months at a fair saving over the cost of purchased power, it would appear that the extra steam available during the heating season, as required for space heating, will further reduce the cost and show a greater yearly saving.

STEAM LOAD CURVE

The steam load curve of this plant for January 4 and 15, 1929, is shown by the dashed line at the top of chart, Fig. 6. The boiler equipment at this plant consists of one 450 and four 308 hp boilers (nominal rating). All boilers are equipped with stokers and forced draft, the stoker equipment being capable of operating the boilers at 200 per cent rating. The boilers are designed for a pressure of 175 lb per square inch gage and are not equipped with superheaters. High pressure steam for certain operations is required at the average rate of approximately 9000 lb per hour, and is fairly constant. The remainder of the steam generated, or 49000 lb average per hour, it is assumed, will satisfactorily supply the remainder of the requirements (cookers, evaporators, drying, water heating, etc.) with a back pressure at the turbines of 20 lb per square inch gage. In order to supply dry steam at the exhaust outlet of the turbines it is proposed to install superheaters for all of the existing boilers. The cost of 13000 Btu coal in the bunkers is $3.75 per ton. The average *boiler and grate* efficiency of this plant is 68 per cent. From an inspection of the electric power load curve it is evident that the full load turbine rating required should be approximately 1500 kw to provide for the peak load. Probably the most practical combination, all things considered, would be the installation of three 750 kw machines, one of which would be a spare unit.

Assuming an initial throttle pressure at the turbine of 160 lb per square inch gage (175 lb absolute), 150 deg superheat and 15 lb drop through throttle with 20 lb per square inch gage (35 lb absolute) back pressure, the ideal water rate with 160 lb absolute pressure is:

$W\,R_{kw} = 29.2 \times 0.879$ (superheat correction) $= 25.5$ lb per kilowatt hour.

Assuming an efficiency ratio ($\varnothing_t$) for the turbine of 0.63 and a generator efficiency of 0.944, the expected full load water rate for a 750 kw turbine-generator set for the assumed condition is:

$$E\,W\,R_{kw} = \frac{25.5}{0.63 \times 0.944} = 42.9 \text{ lb per kilowatt hour.}$$

A guaranteed water rate approximately 10 per cent lower than this figure can be obtained from some manufacturers.

TABLE 6

Load kw	Fractional Load x	Generator Efficiency $\varnothing_x$	Ratio—Generator Efficiency Full Load / Generator Efficiency Fractional Load	Steam per Hour from Willans Line—Lb S	Corrected Total Steam per Hour Lb	Water Rate kw Hour Lb E W Rkw
1500	1.00	0.944	1.000	64350 (S_1)	64350	42.9
1125	0.75	0.931	1.014	51800	52525	46.4
750	0.50	0.909	1.038	39500	41001	54.6
500	0.33⅓	0.890	1.060	31500	33390	66.8

The total estimated steam required at *full load* for two 750 kw units is:

$$2 \times 750 \times 42.9 = 64350 \text{ lb per hour}$$

It is assumed that the ratio *no load* to *full load* steam is $y = 0.22$, the estimated steam at *no load* being equal to 0.22×64350 or 14157 lb per hour. The Willans line *a-b* may now be constructed as previously indicated and is evidently based on the assumption that the generator efficiency remains constant for all loads. The corrected total steam per hour, line *c-d,* is shown in the diagram directly below the Willans line and was plotted from the calculated data given in Table 6, which in turn are based on equation (4).

The corrected total steam per hour for the various loads shown was obtained by multiplying the values as determined from the Willans line by the ratio of the generator efficiency at full load to the generator efficiency at the fractional load.

Any point, as *g* on the electric power load, may be translated into equivalent steam load by simply projecting over to the intersection with the total steam per hour line and transferring the equivalent power load to the steam chart, as indicated at *h.* Any point on the available steam chart curve may be translated into equivalent electric power by simply reversing this process. The deficiency in the process steam at various times during the day for generating the electric power required is shown by the shaded areas on the Process Steam Chart. Integrating these areas the average deficiency in process steam to generate the power required is found to be 5600 lb per hour.

The average process steam available is 49000 lb per hour as determined by integrating the area under the process steam load curve. The average steam generated by the boilers for all purposes is 58000 lb per hour and the average required, if all power were to be generated at the plant, would be $58000 + 5600 = 63600$ lb per hour. The hourly quantities of steam as would be required, if all the power was to be generated, is shown by the steam load curve chart at the top of the figure. This chart shows that the peak load on the boiler plant would remain at 78000 lb per hour, and is in no way affected by the problem of power generation.

The factor of evaporation for 160 lb gage dry saturated steam and 200 deg feed water is 1.0592 and for the same pressure condition with 150 deg initial superheat is 1.148.

The extra fuel required per hour, if all the power is to be generated, is, therefore:

$$\frac{[(63600 \times 1.148) - (58000 \times 1.059)] \times 971.7}{0.68 \times 13000} = 1274 \text{ lb}$$

or

$$\frac{1274 \times 24}{2000} = 15.28 \text{ tons per day}$$

The exhaust from the turbine will have a superheat of approximately 51 deg (Mollier Diagram Fig. 5). Dry steam will therefore be supplied by the turbines to the process.

The deficiency in process steam could be practically eliminated if the full load water rate of the turbines employed could be reduced to 37 lb per kilowatt hour. This would require an ideal water rate ($W\ R_{kw}$) of 21.9 lb, which corresponds to an initial pressure of 235 lb absolute or 220 lb gage. This

increase in pressure would require a new boiler installation, which is not possible in this plant.

The estimated cost of the additional equipment required, if all the power is to be generated at the plant, is as follows:

3—750 kw steam turbine generating sets	$51,000
3—Foundations for above machines	1,500
Superheater	8,500
Piping and electric wiring	17,000
Turbine house	12,000
Total	$90,000

The daily cost of producing electric power based on the preceding data is estimated as follows:

Fixed charges (15% of $90,000 ÷ 365)	$ 37.01
Additional fuel required, 15.28 tons at $3.75	57.30
Additional labor (stationary engineers)	15.00
Additional supplies	4.00
Total	$113.31

The average electric power load for the day was 1138 kilowatts. The estimated cost per kilowatt hour to produce this amount of power is therefore:

$$\frac{113.31}{24 \times 1138} = \$.00415 \text{ for June 4-5}$$

The cost of purchased power at this plant averages $0.007 per kilowatt hour. In order to obtain a fairly correct average for one year it is recommended that the steam and power load charts for not less than 7 representative working days and one Sunday for each month be examined and the deficiency in fuel determined as previously indicated.

For the example given the average hourly deficiency for the year (8760 hours) was found to be somewhat less than for June 4-5, as shown by Fig. 6.

The amount of extra fuel, as previously estimated, may be reduced by the employment of extraction or bleeder type turbines, as later more fully described, by passing the extra steam as required to generate the deficiency in power due to the lack of sufficient process steam through more turbine pressure stages to a final lower back pressure.

It is evident that the extra steam required must be wasted so that any reduction in the amount results in a direct saving in fuel.

It is considerably more tedious to plot the equivalent steam chart, when a bleeder turbine is under consideration, to determine the deficiency in process steam. This is due to the fact that an extraction line as later shown, is required corresponding to each amount of steam it is desired to transfer into electric power or vice versa.

The following method is only a rough approximation, but is sometimes employed for preliminary estimates:

Assuming that the extra steam required is expanded in the turbine down to a back pressure of 17 lb absolute (2 lb gage) the heat available (Mollier diagram) is 187 Btu per pound. The heat available for 35 lb absolute back pressure, as previously determined, is 133 Btu per lb. Therefore $1 - \frac{133}{187}$ or 0.25 (approximately) of the excess steam, as previously calculated (5600 lb

per hour), may be saved by the use of bleeder type turbines, or $0.30 \times 5600 =$ 1680 lb per hour (average) and an estimated saving in fuel of 2.5 tons of coal per day. The saving in fuel per year is $2.5 \times 365 \times \$3.75$ or $3439.00. The extra cost of the bleeder turbine equipment is approximately $10,000.00. The fixed charges on this extra cost are $1,500.00, so that the estimated net accrued saving is a comparatively small amount ($1,939.00).

Extraction or Bleeder Turbines

(Figs. 7, 8) Steam turbines designed to permit the extraction or bleeding of steam from one or more of the pressure stages are known as extraction or

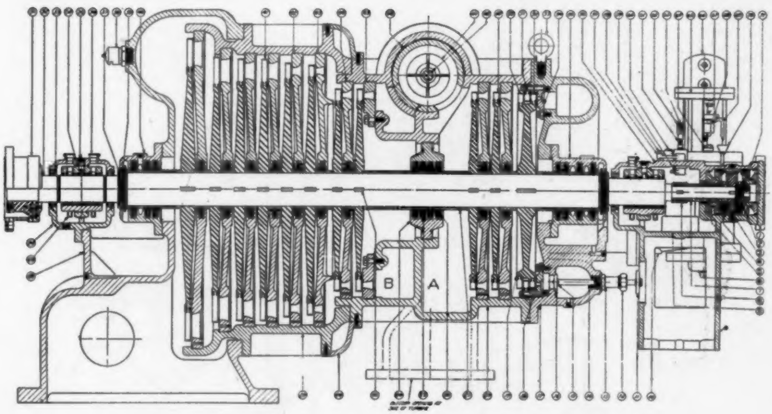


Fig. 7. Cross Section of Moore Extraction Turbine (11 Stages)

bleeder turbines. The turbine may be designed for either condensing or non-condensing operation. This turbine is principally employed for supplying power in industrial plants.

Practically all manufacturing plants employ some steam for process work and generally always for space heating. The pressure required for the large percentage of steam for these purposes generally does not exceed 40 lb per square inch gage. The boilers are generally operated at pressures below 225 lb per square inch when power is to be generated, although in some recent installations 400 lb pressure is employed. This high pressure is frequently required to effect a heat-power balance. Steam at one or more pressure stages of the turbine is bled to supply the process and the remainder passed through additional stages to a condenser or to the parts of the process requiring the lowest pressure and operating as a non-condensing machine.

Condensing operation is often found necessary to secure the amount of power required due to an insufficient supply of process steam at the process pressure or pressures. It is evident, in this case, that more steam must be generated to supply the deficiency, as was shown to be the case in the previous example, and that less extra fuel will be required if the machine is operated condensing.

Bleeder turbines are equipped with an automatic bleeder valve, which admits steam to the low pressure end of the turbine and maintains the bleeder pressure constant. The bleeder valve opens the low pressure steam port or ports with an increasing area as the bleeder pressure rises due to a decreasing demand for process steam, passing the extra steam as may be required for power generation to the low pressure stages of the turbine. An economical application of the bleeder turbine operating condensing is that of supplying hot water for a process which requires both steam and hot water.

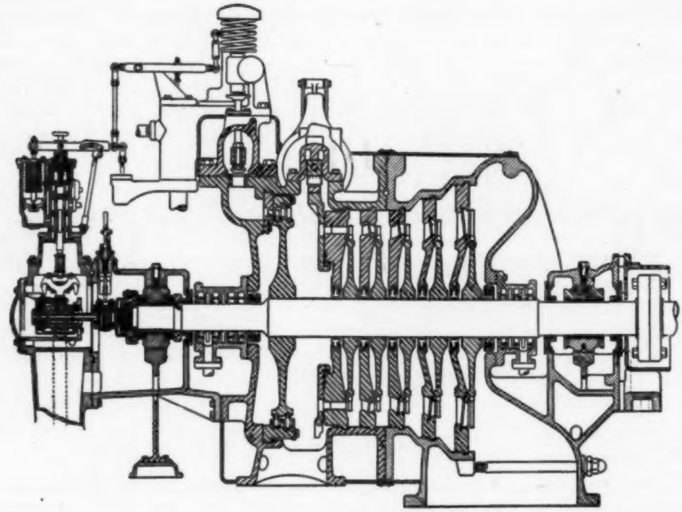


FIG. 8. CROSS SECTION OF GENERAL ELECTRIC COMPANY EXTRACTION TURBINE (6 STAGES)

WATER HEATING FOR PROCESS WORK

When a considerable percentage of the total process steam is required for heating water, the most economical method of obtaining this hot water is from the condenser hot well of a bleeder or extraction type turbine. The bleeder steam is extracted or drawn from the turbine at the pressure stage desired, while the steam required for the water heating is passed on through the low pressure stages of the turbine to the condenser. It is quite evident that more power may be generated in this manner for a given weight of steam employed for water heating due to the considerable difference in water rate between non-condensing and condensing operation. The additional percentage of electric power that may be generated depends on the partial vacuum, the partial vacuum in turn being dependent upon the required temperature of the process hot water.

For all practical purposes the same weight of steam will be required to heat

a given weight of water either at the extraction pressure or the condensing pressure. The temperature of the condenser hot well water is assumed the same as the temperature of the hot water employed in the process.

There exists a difference in temperature between the temperature of the steam corresponding to the condenser vacuum and the hot well temperature (*terminal difference*) from 5 deg to 15 deg, so that the temperature corresponding to the partial vacuum will be approximately 10 deg higher than the hot well temperature. The absolute pressure corresponding to this temperature is, of course, the terminal pressure for the turbine.

The regulation of the high pressure end may be accomplished by either automatically cutting in or out a series of valves controlled by a governor, thus

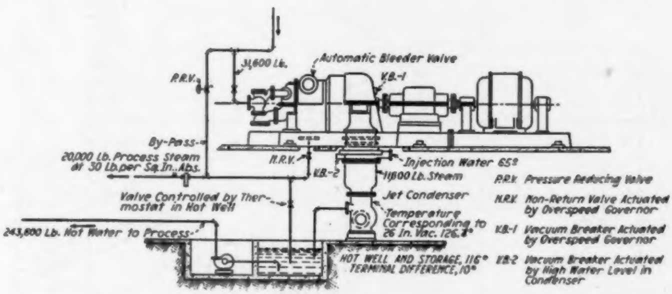


FIG. 9. EXTRACTION TURBINE OPERATING CONDENSING SUPPLYING STEAM AND HOT WATER FOR PROCESS

reducing the loss due to throttling to a minimum or by the action of a single balanced throttle valve controlled by a governor.

The admission of steam to the low pressure end is under the control of a pressure regulator on the extraction line which in turn operates a single throttle valve (bleeder valve) to maintain a constant bleeder pressure (p_e). This valve when partially open produces a throttling action on the steam, and the initial pressure (p_x) to the low pressure end will be less than the constant extraction pressure (p_e) for partial loads. The high pressure end must be capable of carrying the full load with all extraction (no steam) to the low pressure end. This condition gives the maximum amount of steam that will pass through the high pressure end. The maximum amount of steam that will be required to be passed through the low pressure end will be the amount required to develop full load with no extraction, the bleeder valve being wide open. With some extraction, which is the condition of normal operation, it is evident that the bleeder valve will only be partially open and throttling will take place from p_e to p_x.

The following formulae may be employed in solving for the total steam to be supplied the turbine to develop full load with and without extraction:

With all extraction.

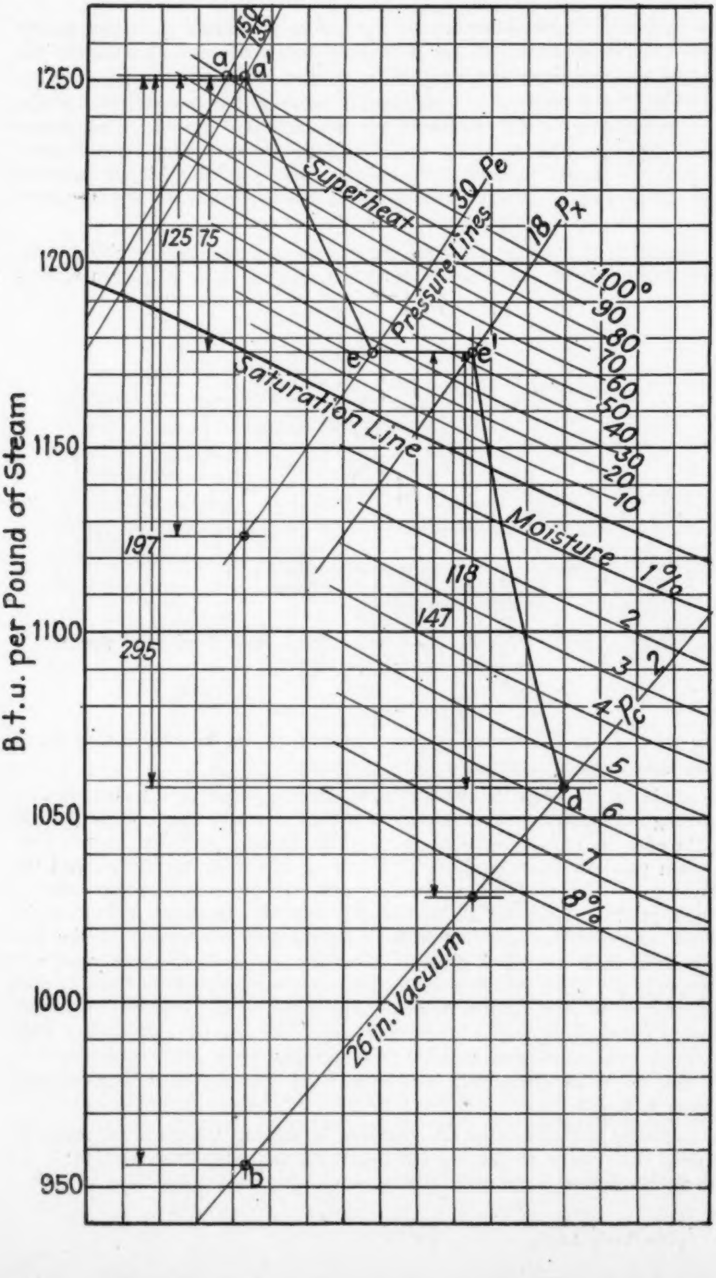


FIG. 10. CONDITION LINE SHOWN ON MOLLIER DIAGRAM FOR 20,000 LB EXTRACTION PER HOUR

$$S_m\ h_1 = \frac{3412 \times N}{\varnothing_m \times \varnothing_g} \tag{1}$$

With no extraction.

$$S_x\ h = \frac{3412 \times N}{\varnothing_m \times \varnothing_g} \tag{2}$$

With extraction—(W lb per hour)

$$S\ h_1\ (S—W)\ h_2 = \frac{3412\ N}{\varnothing_m \times \varnothing_g} \tag{3}$$

$$\text{also; } \frac{p_x}{S—W} = \frac{p_e}{S_x} \tag{4}$$

Where—

H=theoretical adiabatic heat drop for complete turbine between the initial pressure and condenser pressure, Btu per pound

h_{a1}=theoretical adiabatic heat drop for high pressure end of turbine between the initial pressure and extraction pressure, Btu per pound

h_{a2}=theoretical adiabatic heat drop low pressure end of turbine between the initial pressure for low pressure end and condenser pressure, Btu per pound

$\varnothing_1$=internal efficiency of complete turbine with no extraction

$\varnothing_{a1}$=internal efficiency for high pressure end

$\varnothing_a{}^2$=internal efficiency for low pressure end

$h=\varnothing_1\ H$=heat absorbed by rotors complete turbine, Btu per pound

$h_1=\varnothing_{a1}\ h_{a1}$=heat absorbed by high pressure end, Btu per pound

$h_2=\varnothing_{a2}\ h_{a2}$=heat absorbed by low pressure end, Btu per pound

N=kilowatts delivered at generator terminal

3412=Btu equivalent of one kilowatt

S=steam supplied turbine with extraction, pounds per hour

W=steam extracted per hour, pounds

$S—W$=maximum steam through low pressure end, pounds per hour

S_m=maximum steam through high pressure end (all extraction), pounds per hour

S_x=maximum steam through low pressure end (no extraction), pounds per hour

p_e=extraction pressure, pounds sq in. absolute

p_x=initial pressure low pressure end, pounds sq in. absolute

Equations (3) and (4) must be solved tentatively employing the Mollier chart.

Example. Let it be required to determine the total weight of steam (S) to be supplied a 1000 kw extraction turbine to develop full load with 20,000 lb of steam extracted per hour (W) at an absolute pressure p_e = 30 lb per square inch and also the weight of water that may be heated per hour from an initial temperature of 65 F to 116 F. Also construct a Willans line steam chart (Fig. 11) and show the condition line on a Mollier chart (Fig. 10). The initial pressure at turbine throttle assumed is 150 lb absolute with 100 deg superheat. A pressure drop of 15 lb per square inch will be assumed through the main throttle valve.

Assuming a 10-deg *terminal difference,* the final steam temperature will be 116 + 10 or 126 F which corresponds to a condenser pressure $p_c = 2$ lb absolute or 26 in. vacuum referred to a 30 in. barometric pressure.

The Willans lines for the high pressure end of the turbine and for the complete turbine (Fig. 11) will be the first constructed by assuming the following data: Internal efficiency for the high pressure end and when developing full load (1000 kw) with all extraction assumed as $\varnothing_{s_1} = 0.63$. The efficiency for the complete turbine operating condensing is assumed as $\varnothing_t = \varnothing_1 \times \varnothing_m = 0.64$ when turbine is developing full load with no extraction and 26 in. vacuum. Assuming $\varnothing_m = 0.95$ then $\varnothing_2 = 0.67$. The heat available from the Mollier diagram (Fig. 10) for adiabatic expansion between the initial pressure (Fig. 10) for adiabatic expansion between the initial pressure (135 lb) and extraction pressure (30 lb) is 125 Btu and between the initial pressure and condenser pressure

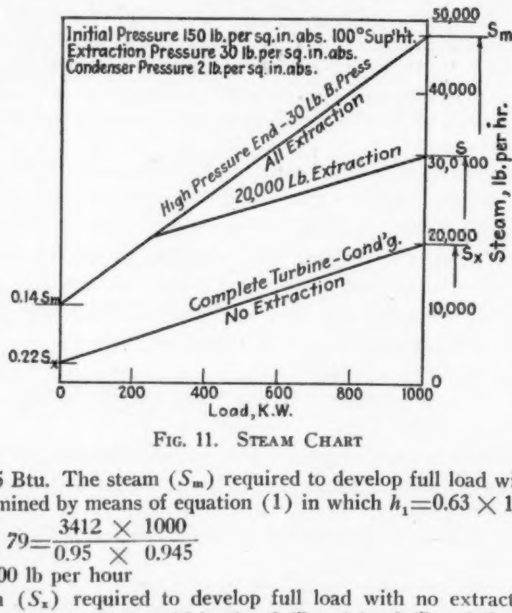


FIG. 11. STEAM CHART

(2 lb) is 295 Btu. The steam (S_m) required to develop full load with all extraction is determined by means of equation (1) in which $h_1 = 0.63 \times 125$ or 79 Btu.

Then $S_m \times 79 = \dfrac{3412 \times 1000}{0.95 \times 0.945}$

or $S_m = 48{,}000$ lb per hour

The steam (S_x) required to develop full load with no extraction is determined from equation (2) in which $\varnothing_t = 0.67$ and $h = 0.67 \times 295 = 197$.

$S_x \times 197 = \dfrac{3412 \times 1000}{0.95 \times 0.945}$ or $S_x = 19300$ lb per hour

The no-load steam for the high pressure end is assumed as $0.22S_m$ non-condensing operation and for the complete turbine as $0.14S_c$ condensing operation. The Willans line for the high pressure end and for the complete turbine are then drawn on the steam chart (Fig. 11) as shown.

In order to construct the extraction line for 20,000 lb extraction per hour it is first necessary to assume the internal or stage efficiencies for the high and low pressure ends. These will be assumed as $\varnothing_{s_1} = 0.60$ and $\varnothing_{s_2} = 0.80$.

A throttling action takes place with extraction and it is necessary, as previously mentioned, to solve equations (3) and (4) tentatively by assuming various values for p_x and determining corresponding values for h_2 by means of the Mollier chart until both equations (3) and (4) are satisfied. Assume for example $p_x=18$ with $s_x=19300$ from equation (2), then equation (4) gives $S=31580$.

The heat available between $p_x=18$ and the condenser pressure $p_e=2$ is, $h_{as}=147$. Therefore, $h_2=0.80 \times 147=118$. Substituting this value in equation (3): $S75$ $(S—20{,}000)$ $118=\dfrac{3412 \times 1000}{0.95 \times 0.945}$ or $S=31740$, which is a sufficiently close agreement with the value previously determined by equation (4).

Other points on the extraction curve may be determined by assuming values for p_x solving for S_x in equation (4) and then substituting the values of S_x and h in equation (3) and solving for the corresponding values of N.

The results of these calculations are given in the following table:

Load kw N	p_x	S	h_{a2}	h_2	$S—W$	Weight of Water Heated per Hour
1000	20	31600	147	118	11600	232000
843	15	29650	135	108	9650	193000
670	10	26433	109	87	6433	128660
543	6	23860	76	61	3860	77200

A smooth curve drawn through the points located by values of N and S on the steam chart is the 20,000 lb extraction curve shown. Extraction lines for the other amounts of extraction are determined in a similar manner. For all practical purposes a straight line may be substituted for the curved line. The weight of water that may be heated from 65 F to 116 F assuming that the latent heat as constant at 1020 Btu per pound is $\dfrac{(S—W)\ 1020}{116—65} = 234{,}000$ lb per hour and is evidently the weight of condensing water required per hour at an initial temperature of 65 F. The condition line drawn on the Mollier diagram for the turbine when developing full load with 20,000 lb extraction is shown as *a'ee'd*, Fig. 10. The extraction shown (Fig. 11) is based on a constant efficiency of generator for all loads. If corrected for a decrease in efficiency for partial loads the line becomes somewhat steeper.

TWO STAGE WATER HEATING

The previous example illustrates single stage water heating. A more economical method (Fig. 12) is to use two or more stages, employing a lower partial vacuum for the first stage and complete the heating by steam bled at a somewhat higher pressure. Assume the same initial pressure as for the previous problem with 15 lb per square inch drop through throttle and 69240 lb water per hour to be heated from 65 deg to 200 deg. Dividing the temperature rise (135 deg) equally between two stages, a rise of 67.5 deg for each stage is obtained.

The steam required to heat the water is:

$\dfrac{69240 \times 135}{1000}=9348$ lb per hour or 4674 lb per stage. The temperature of the

hot well for the final or second stage is 200 deg and with a 10 deg terminal difference gives 210 deg for the steam temperature to condenser, which corresponds to 14.1 lb per square inch absolute pressure. The temperature of the hot well for the first stage is 200 deg and with a 10 deg terminal difference gives 210 deg for the steam temperature to condenser, which corresponds to 14.1 lb per square inch absolute pressure. The temperature of the hot well for the first stage is 200—67.5=132.5 deg. This is also the temperature of the injection water for the second stage condenser.

With a 10 deg terminal difference the steam temperature for the first stage is 132.5+10=142.5 deg, corresponding to a terminal pressure for the turbine of 3.1 lb per square inch absolute. The theoretical water rate for the com-

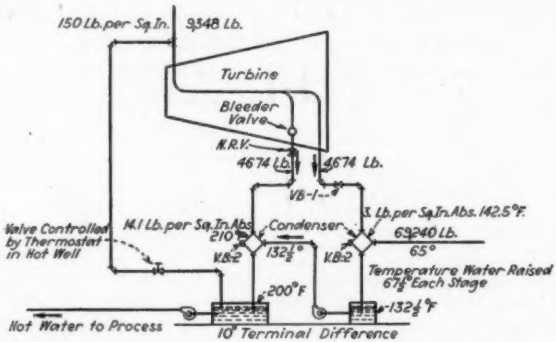


FIG. 12. EXTRACTION TURBINE—TWO STAGE WATER HEATING

plete turbine is 12.4 lb per kilowatt hour and for the part of the turbine delivering steam at 14.1 lb per square inch is 19.3 lb per kilowatt hour. Assuming overall efficiencies of 0.60 and 0.64 the expected water rates become $\frac{19.3}{60}=32.1$ (2nd stage heating) and $\frac{12.4}{64}=19.4$ for complete turbine (1st stage heating).

The approximate power that could be generated by two stage heating would be:

Steam bled at

14.1 lb sq in.=4674 divided by 32.1=145.6 kw

Steam condenser at

3.1 lb sq in.=4674 divided by 19.4=240.9 kw

Total ....9348.......................................386.5 kw

If the single stage heating was employed the power generated would be:

$$\frac{9348}{32.1}=291.2 \text{ kw}$$

The gain by two stage heating is therefore:

386.5—291.2=95.3 kw or approximately 33 per cent over the single stage method.

EXHAUST STEAM SPACE HEATING—STEAM VERSUS HOT WATER SYSTEMS

When turbine exhaust steam is to be employed for space heating considerably more electrical power may be generated with a hot water system than with a vacuum steam system on account of the fact that a lower back pressure may be carried at the turbine as will be evident by the following example: (Fig. 13).

Steam System—Assume a heating load of 100000 sq ft of equivalent direct steam radiation or 24276000 Btu per hour or 24276 lb of steam per hour (latent heat assumed as 1000 Btu per pound). Initial steam pressure 200 lb absolute, no superheat and 20 lb absolute (5 lb gage) back pressure.

The ideal water rate for the turbine $WR_{kw}=20$. The expected water rate

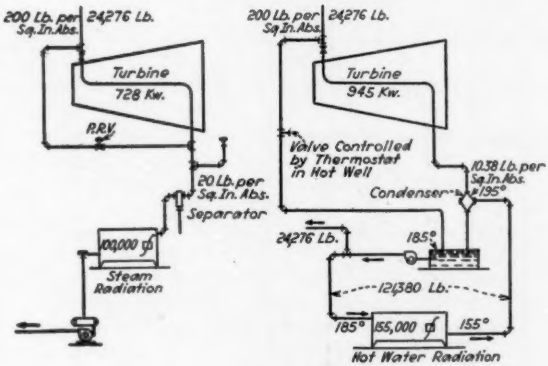


FIG. 13. EXHAUST STEAM HEATING

for an assumed combined efficiency of turbine and generator of 60 per cent is $\frac{20}{60}=33.3$ lb per kilowatt hour. The power that could be generated with the vacuum steam system is therefore $\frac{24276}{33.3}=728$ kw.

Hot Water System—Assuming an initial water temperature of 185 deg and final temperature of 165 deg for the radiation, the amount of water to be circulated is: $\frac{24276000}{185\text{-}165}=1213800$ lb per hour.

The initial temperature of the injection water will be 165 deg and the final temperature 185 deg. The steam temperature for a 10 deg terminal difference is 185+10 or 195 deg corresponding to an absolute pressure of 10.38 lb per square inch. Assuming a combined efficiency of generator and turbine of 63 per cent the expected water rate $(E\ W\ R_{kw})$ is $\frac{16.2}{0.63}$ or 25.7 lb per kilowatt hour.

The power that could be generated for the assumed conditions is, $\frac{24276}{25.7}=$

945 kw. The increased power that could be developed is, 945-728 or 217 kw or a gain of approximately 30 per cent in power with the hot water system. The two systems would, of course, theoretically give precisely the same results as to power developed and fuel economy if the steam system was provided with an air and vacuum pump of sufficient capacity to produce the partial vacuum as stated for the hot water system.

The diagram on the right of Fig. 14 gives the kilowatts generated per thousand pound of dry saturated steam supplied the turbine at various initial and terminal pressures. When the steam is initially superheated divide the values as determined by the superheat factors (Fig. 3). The expected power

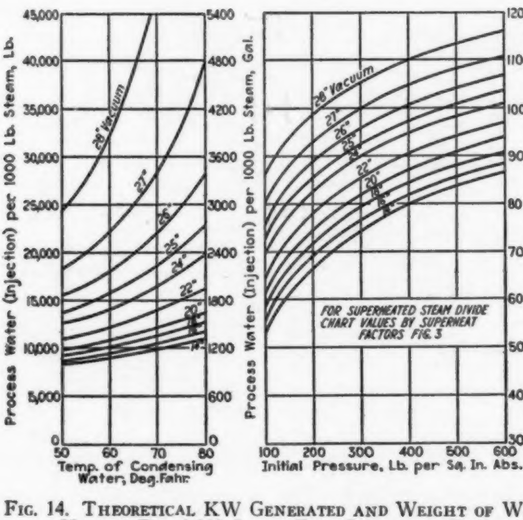


FIG. 14. THEORETICAL KW GENERATED AND WEIGHT OF WATER HEATED PER 1,000 LB OF DRY SATURATED STEAM

per 1000 lb of steam is determined by multiplying the diagram values (corrected for superheat if any) by the combined efficiency of the turbine and generator (Table 5).

The amount of water which can be heated per 1000 lb of steam supplied which is also equal to the amount of injection water required at a temperature of 10 deg less (terminal difference) than the temperature corresponding to the partial vacuum may be read direct from the left chart. See Fig. 3 for temperature—partial vacuum diagram. The heat to be extracted per pound of exhaust steam is here assumed constant at 1000 Btu, a fair average value when superheated steam is supplied to the turbine.

The following references will be found useful to those interested in the study of this subject:

1. The Bleeder Turbine—Its History and Theory, by J. L. Moore, *Trans. A. S. H. V. E.* 1922.

2. Stage Feed Water Heating, by E. H. Brown and M. K. Drewry, *Trans. A. S. M. E.* 1923.
3. The Value of High Pressures in Industrial Plants, by Wm. F. Ryan, *Trans. A. S. M. E.* 1925.
4. Steam Power Plant Engineering, by L. A. Harding—Published by John Wiley and Sons, Inc.
5. Balancing Heat and Power in Industrial Plants, by R. V. Kleinschmidt with discussion by various engineers. *Trans. A. S. M. E.* 1929.
6. Heating and Process Loads, by A. R. Acheson, Heating, Piping and Air Conditioning, July, 1929.
7. Steam Turbine Performance, by A. G. Christie, Kent's Mechanical Engineers Handbook.
8. The Application of Low Pressure Turbines, by Francis Hodgkinson, bulletin published by the Westinghouse Machine Co.
9. Mixed Pressure Turbines, by E. D. Dickson, General Electric Review.
10. Steam and Gas Turbines, by Stodola and Lowenstein.
11. Characteristics of Bleeder Turbine, by R. B. Walden, *Power,* 1928.
12. Byproduct Generation in District Heating Plant, by J. H. Walker, *Power,* 1928.
13. Bleeder Turbines, by M. D. Church, *Power,* June 1929.
14. Multi-Stage Water Heating, by M. D. Church, *Power,* July, 1929.
15. The Ruths Steam Accumulator by R. A. Langworthy. *Trans. A. S. M. E.* 1929.

The author is indebted to L. A. Cundall for assistance rendered in calculating Tables 1 to 4 and the Chart Fig. 3.

DISCUSSION

Perry West: I should like to ask the author what allowance should be made between the theoretical water rate and the practical rate.

J. W. Meyer, Jr.: I should like to caution on one point which Mr. Harding did bring out very forcefully, and that is the study of the cycle of operation of the heating and the electric load. There are two extremes. We have the condition where the steam load and the electric loads are coincident. We have the other extreme in which there is a lack of coincidence, in which the steam must be exhausted to the atmosphere. Somewhere in between those two points rests each particular case, and the results to be obtained depend upon the accuracy with which you determine the coincidence of the two loads. It has been our own experience that in a number of cases where such studies have been made, the after-effect or the result obtained, was far from the result obtained from the original analysis and it was very largely due to failure to take into consideration this cycle of operation.

Webster Tallmadge: Having been in turbine work for a number of years it comes to mind that there are two points that might be considered. First, if you do have to plot a consumption and demand curve, do it by weeks consecutively throughout the year rather than months, on account of the fact that there are different days in the month that very frequently distort the figures.

Another thing is that for a given efficiency the peak load demand on a plant determines the capitalization, very frequently by storing heat in water used for manufacturing processes in off periods of live steam demand. By this method the steam demand will even out the peak load because the high water demand and the steam demand do not pull steam together. Very often on such as bleacheries the capitalization of steam generating equipment will be lower by a little planning and, therefore, reduce the operating overhead.

F. D. Mensing: My first caution is one you should all consider. No man

can walk into a power plant and do justice to his client by working on data gathered over a short period of time. To those who take up this branch of engineering, I would suggest that instead of going after jobs they go after clients. You can generally justify the installation of recording instruments and make enough of a saving resulting from those instruments to justify their installation. Put off the time of power house change if possible for a year. Then work up your charts for the year. The average instruments for the boiler room can be installed for about $2,000. They can be used afterward. You can put electrical instruments in there, both recording and integrating type, that give you a chart, for somewhere around five or six hundred dollars.

We, in this climate, for instance, have an average mean temperature on the 22nd day of January of 31 F. Exactly six months later we have an average mean of 76 F. A wonderful climate on the average means but diabolical on peaks. We go up to 106 and down to 6 below. So you are bound to get overlaps.

After you have accumulated your year's charts, which you can do automatically as they send them in to you, you can plot those on paper which is available, by days, months, and years, and then you have a cold-blooded proposition that you can walk into a client and say "This is what is taking place; this is what should have taken place." You can pick out any day of the year, tell him what his heating load is going to be in pounds of coal or in dollars. You can tell him what his electrical load is going to be at any particular time. If you plot production, which you can get from them, you can predict 5 years ahead with surprising accuracy. Now this is not something picked out of the atmosphere. It is something my office often goes through, something that has just been gone through in the last few days. It meant a new power plant and deciding whether that plant would be in Philadelphia or elsewhere.

A. M. HUSTOEL: I would like to correct an impression perhaps Mr. Harding and some of the other engineers have. He said that the turbine manufacturers are sick and tired perhaps of giving them the data. I can speak for one turbine manufacturer: we would be glad to give you the information especially if you prepare your data as carefully as Mr. Harding does, and we would like to help you solve these problems. The matter of water rates is not as definite as you seem to think and we have to know their problem to help them. If you will ask us, I am sure we will always be ready to submit you all the data necessary for your problems.

L. A. HARDING: In reply to Mr. West with regard to the difference between the ideal or theoretical water rates and the expected water rates, if you will refer to Table 5 you will find approximate full-load efficiencies, multi-stage impulse turbines, geared type. The geared type is the one that is largely used The difference in efficiency between the geared and direct connected job is very little. The efficiency of the gears is quite high, running in the neighborhood of 97 and 98 per cent. Generator efficiencies are given by the third column and the combined efficiencies in the last column.

Mr. Mensing calls attention to the fact that the employment of steam flow meters and watt meters is essential in an investigation of this sort; the employment of averages for the day or month will almost invariably result in conclusions which are misleading and of no value where one is dealing with both a varying steam and electric load.

No. 850

PRESSURE DIFFERENCE ACROSS WINDOWS IN RELATION TO WIND VELOCITY

By J. E. Emswiler,[1] Ann Arbor, *and* W. C. Randall,[2] Detroit, Mich.

MEMBERS

THE amount of air passing through a window as leakage is dependent upon the pressure difference across the window, and the crack opening. At the present time, there is a considerable mass of data from laboratory research showing what the leakage is for various kinds of windows for any specified pressure difference; but there is little definitely known about what the pressure difference actually is or is likely to be in any given building. In the absence of knowledge regarding this quantity. it is usual to assume a certain wind velocity appropriate to a given locality, and take the leakage as that corresponding thereto, which is equivalent to assuming a certain pressure difference at the window.[3]

Tables showing the relationship between chosen wind velocities and window leakage per foot of crack that may be expected to occur with pressure differences corresponding to those wind velocities, are given on pages 51 to 54 of The Guide 1929 for various types of windows.

The principal object in predetermining window leakage is to enable due allowance to be made in the amount of heating surface for the expected maximum infiltration. This maximum does not mean the leakage that will occur with the highest momentary gust of wind, nor the highest rate for any one hour, but should rather be considered as that which may occur during several hours of the heating season.

It is stated in The Guide that "the heat allowance for infiltration through cracks must be based on the average wind velocity for a given locality." It seems probable that the *average wind velocity,* as the term is used there, does not mean the actual average for all the hours of the heating season. In the case of transmission losses, The Guide suggests 15 mph as an *average wind velocity,* and it is probably implied that a similar value is intended to be used in the calculation of infiltration losses, in the absence of specific data. It is also stated in The Guide that "a further allowance must be made for the

[1] Professor of Mechanical Engineering, University of Michigan.

[2] Chief Engineer, Detroit Steel Products Co.

[3] For standard air density, the formula $p=0.00048M^2$ is used to calculate pressure corresponding to a given wind velocity or vice versa, where p=pressure in inches of water and M=wind velocity in miles per hour.

Presented at the 36th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1930.

direction of the prevailing wind in any locality, which shall be done by adding 15 per cent to the infiltration losses on the sides of the building exposed to the prevailing winds." The tables of THE GUIDE are given in two parts, the values of Part II being 80 per cent of those for Part I (in which leakage as determined from experiments is given), to make allowance for an opposing

FIG. 1. RECORDING GAGE USED TO MEASURE PRESSURE DIFFERENCE ACROSS WINDOWS

pressure built up on the inside as a result of air being forced in by the excess wind pressure on the outside.

In a former paper by the authors,[4] the need for more definite information on the subject of pressure difference across windows was emphasized, and it was suggested that a program of work along this line should be started. It was with this thought in mind that the study described in the present paper has been undertaken. The definite objectives are to attempt to show how the chosen value of wind velocity for THE GUIDE Tables should be rationally determined for a particular case; to ascertain if the factor of 80 per cent of Part II is a reasonable value; and, if possible, to see what is the effect of

[4] The Weathertightness of Rolled Section Steel Windows, TRANSACTIONS, A. S. H. V. E., 1928, p. 527.

temperature difference. It is to be understood that this study is by no means complete, but is offered at this time rather more in the hope of receiving criticisms and suggestions regarding the mode of procedure in the interpretation of data, than to present final facts and conclusions.

Means of Obtaining Data

A record of pressure difference across two windows in one of the buildings of the University of Michigan was obtained continuously over a period of 45 days, extending from the middle of January to the first of March, 1929. A recording pressure gage with a 24-hour chart was used. A photograph of the gage, known as a hydro-recorder, is shown in Fig. 1. A narrow felt-trimmed board, with a hole through it into which a brass tube was pushed, was slipped under the lower sash, and the crack thus opened at the meeting rail was stopped with another felt-trimmed strip. The brass tube was connected to the gage which was located in the room, and the resulting record showed the excess or deficiency of the outside pressure over that inside. The connection of the gage to the window may be seen in the picture. Fig. 2 shows a part of the record during a period when there was but little wind, and Fig. 3, of a part when the wind was rather high, and from the direction of exposure. The highest momentary pressure difference recorded was 0.88 in. of water, which, in terms of wind velocity, would be about 42 mph.

Two windows were chosen, on the west end of the building, one on the second floor, and one on the fourth. There was free exposure to wind from the west and southwest, as illustrated in Fig. 4. The gage was connected at the second floor window for 28 days continuously, and then moved to the fourth floor, where a record for 17 days was obtained.

Anemometer records of wind velocity and direction as well as thermograph records of temperature were available at the University Observatory. The data of these records were transferred to the pressure difference charts as illustrated in Figs. 2 and 3. For convenient reference, the maximum pressure difference, and the average pressure difference as nearly as it could be determined, were read from the chart curve, and these quantities appear as lines 4 and 5. In this manner, all of the necessary data are synchronized on the pressure difference charts, with the graphic record of the pressure difference in direct view above.

Selecting Wind Velocity in Estimating Infiltration Loss

From Table 1, which is a summary of the wind data over approximately the 45-day period before alluded to, it is seen that the average wind velocity is 7.7 mph over the entire time. It is also seen that the wind is predominantly

Table 1. Summary Wind Data

Item	Direction from which wind came								
	N	NE	E	SE	S	SW	W	NW	Total
1. No. of hours........	97	50	129	85	57	155	179	289	1041
2. Miles of wind.......	569	187	892	509	308	1454	1656	2437	8012
3. Average velocity....	5.9	3.7	6.9	6.0	5.4	9.4	9.3	8.9	7.7
4. Per cent of hours...	9.3	4.8	12.4	8.2	5.5	14.9	17.2	27.7	100.0
5. Per cent of wind....	7.1	2.3	11.1	6.5	3.8	18.1	20.7	30.4	100.0

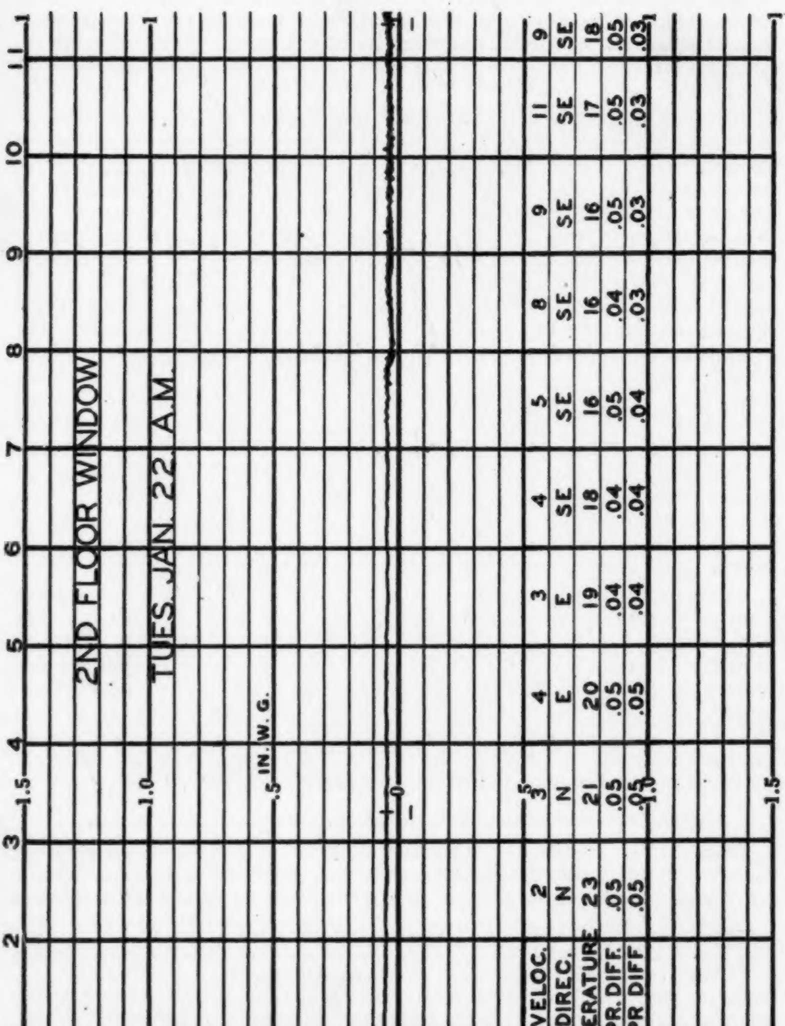


FIG. 2. PORTION OF PRESSURE DIFFERENCE RECORD ACROSS TWO WINDOWS WITH BUT LITTLE WIND VELOCITY

from the SW, W, and NW, nearly 70 per cent of the wind coming from these three directions, and only 30 per cent from the other five.

In order to get at usable maximum values of wind velocity, Table 2 has been prepared, in which the wind is classified as *prevailing* and *non-prevailing*, and the number of hours is shown during which the wind blew at each velocity designated. Separate consideration is given to those hours of the day included between 7 a.m. and 5 p.m., which may be regarded as the ordinary period of occupancy of a building.

From the prevailing directions, the wind had a velocity of 15 mph or more for 73 hours during the month and a half investigated. Counting only the 10 hours per day of ordinary occupancy, the wind had a velocity of 15 mph or more for 25 hours, which is somewhat less than ten twenty-fourths of 73, indicating that most of the higher wind velocities occur at night. If it be assumed that the wind statistics for the other month and a half of the season are like those studied, it could be said that the wind from the prevailing directions has a velocity of 15 mph or more during 50 hours of the time of occupancy of a building. Perhaps this would not be an unreasonable figure upon which to base allowance of heating surface requirements, considering the fact that the periods of high wind velocity will not always occur simultaneously with the periods of lowest temperature. The average speed of the prevailing wind for all those hours during which the velocity is 15 mph or more is found to be about 19 miles. Hence, for this particular case, it would appear that a value of 19, or say 20 mph would represent a reasonable value for wind velocity to be chosen for those sides of a building exposed to the prevailing wind. To obtain more accurate results, wind data for several seasons should be studied in this manner.

Turning now to the non-prevailing winds of Table 2, it is seen that the wind had a velocity of 9 mph or more during 28 hours of the time of occupancy, for the month and a half period, or perhaps 56 hours for the entire season. The average speed of the non-prevailing wind for all those hours during which the velocity is 9 mph or more is found to be about 13 miles.

Having thus determined what is to be considered as the wind velocity, the next objective is to ascertain if possible what is the relation between wind velocity and actual pressure difference, and to find out if the factor of 80 per cent of Part II of THE GUIDE tables is justified.

TABLE 2. PREVAILING AND NON-PREVAILING WINDS

Wind Velocity	Number of Hours			
	Prevailing Wind		Non-Prevailing Wind	
	SW. W. and NW.		N. NE. E. SE. and S.	
	All Hours of Day	7 A. M. to 5 P. M.	All Hours of Day	7 A. M. to 5 P. M.
1. 25 MPH or more	8	5	0	0
2. 20 MPH or more	35	10	0	0
3. 15 MPH or more	73	25	9	1
4. 10 MPH or more	246	78	44	14
5. 9 MPH or more	...	..	69	28

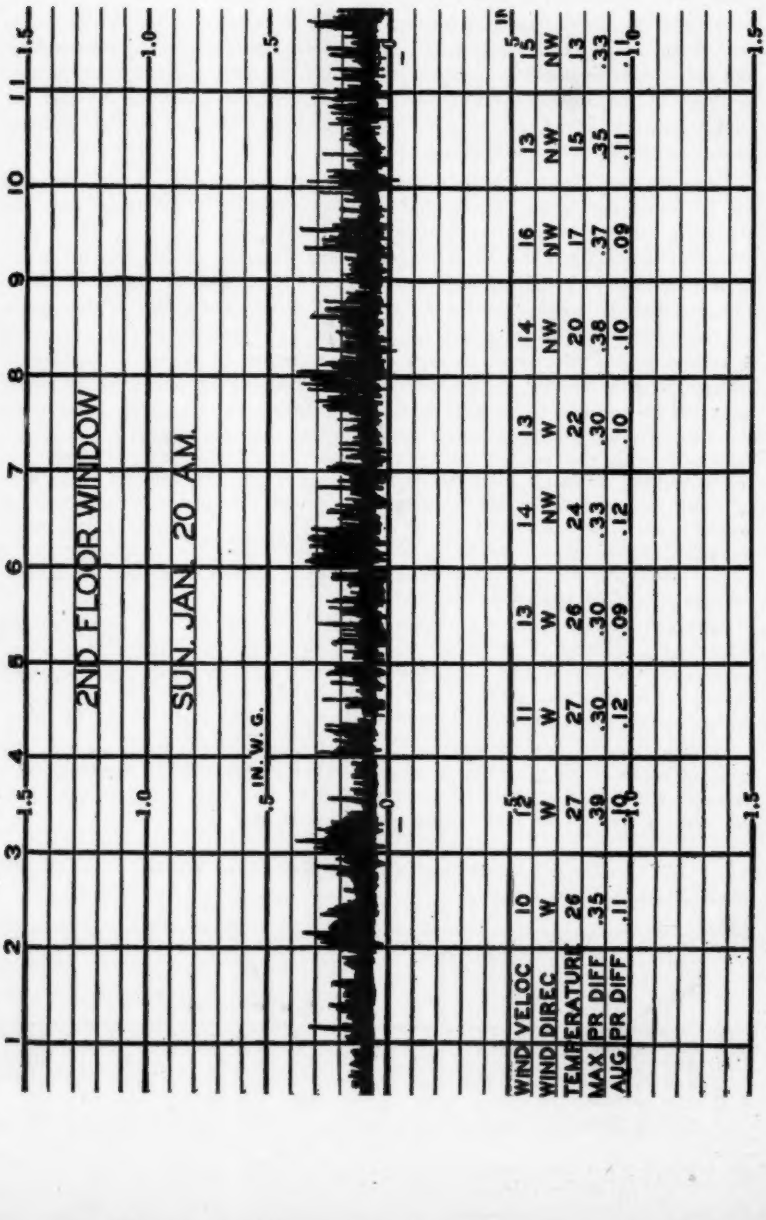


FIG. 3. PORTION OF PRESSURE DIFFERENCE RECORD ACROSS TWO WINDOWS WITH HIGH WIND VELOCITY

RELATION OF PRESSURE DIFFERENCE TO WIND VELOCITY

A record of pressure difference across two windows of a chosen building, one on the second floor and one on the fourth floor immediately above (see Fig. 4) was obtained in the manner heretofore described. Owing to the proximity of an adjacent building, the windows were sheltered from a northwest wind, but were fully exposed to the west and southwest. The relation between pressure difference across the second floor window and the wind velocity from the west or southwest (the direction of exposure of the window), is shown by the curve of Fig. 5. Each point represents the average

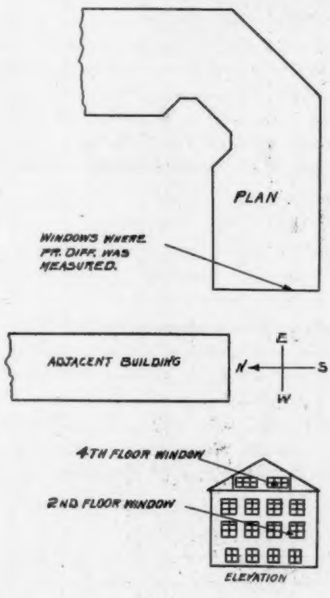
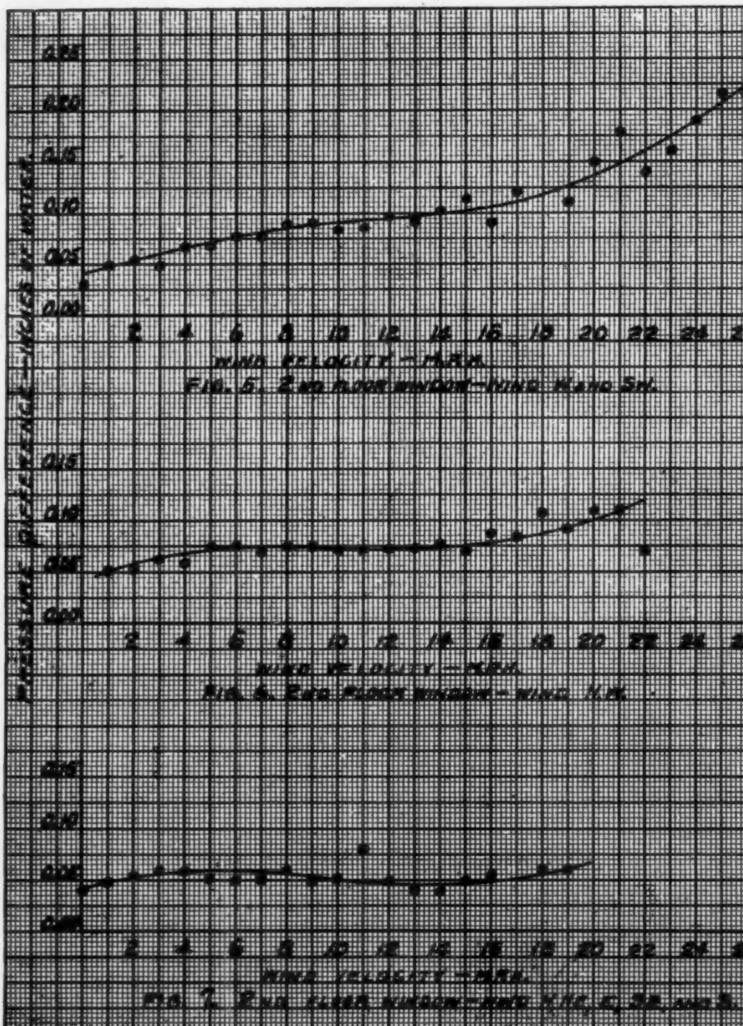


FIG. 4. SHOWING LOCATION OF WINDOWS

of the mean hourly pressure differences for the number of hours during which the wind velocity had the particular value for which the point is plotted. The points lie quite consistently near the smooth curve, except in the region of higher wind velocity, where each one is the average of fewer values than is the case in the region of lower wind velocity.

The curve of Fig. 6 represents the case where the wind came from the northwest, which is considered separately because the window is sheltered in this direction by the adjacent building as shown by Fig. 4. The curve of Fig. 7 shows how the pressure difference on this second floor window is influenced by wind coming from directions other than southwest, west or northwest. The curves of Figs. 8, 9, and 10 present the same kind of picture



FIGS. 5, 6, 7. CURVES SHOWING RELATION BETWEEN PRESSURE DIFFERENCE AND WIND VELOCITY FOR 2ND FLOOR WINDOWS

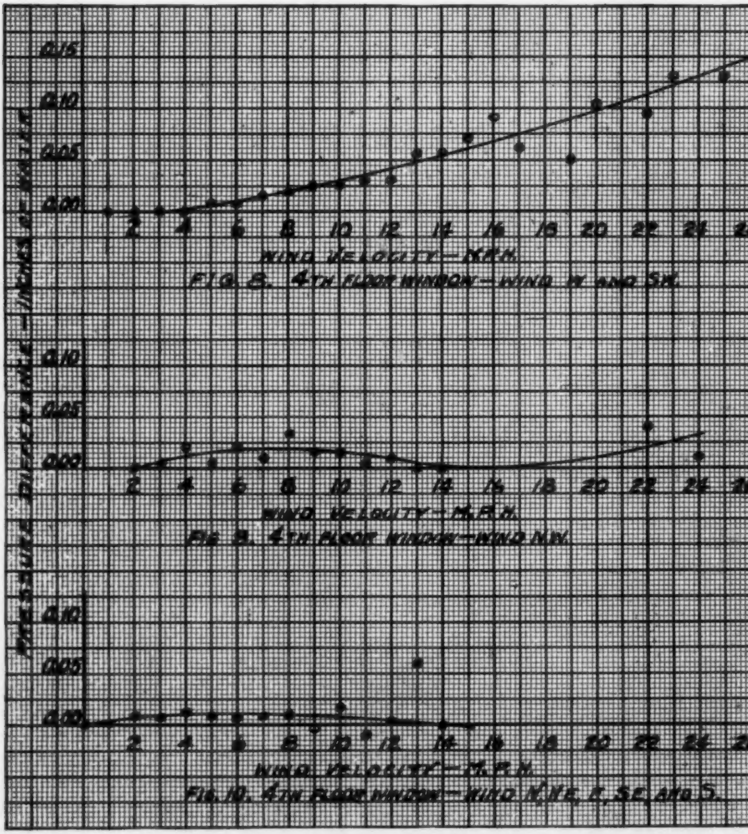
of the relation between pressure differences and wind velocity for the fourth floor window.

It might have been expected, with winds coming over or around the building, from directions other than southwest, west or northwest, that there would have been a tendency to form a slight vacuum at the leeward end in which the windows were located, and thus materially reduce the excess of outside pressure over inside pressure, or even reverse the direction of excess at the upper window. This same effect might even be expected in some degree when the wind came from the northwest, blowing over or around the adjacent building (see Fig. 4). The curves of Figs. 6, 7, 9, and 10, do indeed suggest this very thing, as all of them show a slight drop after the wind attains a velocity of about 8 mph. None of the curves reveal an excess of pressure inside, over that outside at the upper window, as would be anticipated by the theory of effect of temperature difference. However, there is a reason to account for this situation, which will be explained.

Since the principal object of this part of the study is to examine the pressure difference over windows exposed to the wind, attention will now be directed only to the curves of Figs 5 and 8, and these may be considered to represent what would occur at any window about the building when the unobstructed wind blew against it. The pressure differences at the windows at the two levels investigated are compared in Fig. 11, where *A* is the curve of Fig. 5 (second floor window), and *B* is the curve of Fig. 8 (fourth floor window). Curve *C* of Fig. 11 is the computed wind pressure drawn in for reference. It is seen that the curves for the two windows are more or less parallel, *A* being displaced above *B* by an average amount of 0.053 in. of water. Using the average temperatures that prevailed and the height between the windows, which is about 28 ft, the head caused by temperature difference is calculated to be 0.041 in. of water, which should express the amount by which the pressure difference at the lower window is greater than that at the upper, and pretty well accounts for the actual difference of 0.053 in. between the curves. If it were not for the upward bulge in the curve *A* between wind velocities of zero and about 14 mph, for which no reason seems to be evident, it is probable that the difference between the curves *A* and *B* would be almost exactly accounted for by the effect of temperature.

It would be anticipated that with little or no wind, the temperature difference would produce a negative effect at the upper window—that is, the inside pressure would be greater than the outside, and outflow would occur there. This would be evidenced by the curve *B* of Fig. 11 coming below the base line at low wind velocities, which it does not do. It is believed that the presence of gravity flues in the rooms in which the windows are located accounts for the apparent departure from theory. The gravity flue on the second floor drew air from that room, and thus tended to increase the excess of outside pressure. At the same time the loss of air here reduced the amount that would have otherwise naturally ascended to the fourth floor by way of open stairs in other parts of the building, and so relieved the interior pressure on the fourth floor windows. Also the gravity flues in the fourth floor room provided an escape for air which further relieved the inside pressure there, with the net result that there was no excess of inside pressure over that outside on the fourth floor windows. It is probably merely accidental that the pressure difference at the fourth floor happened to be exactly zero with no wind.

In Fig 12, curve *D* is the mean of curves *A* and *B*, of Fig. 11—that is, it represents the average of the pressure differences observed at the second and fourth floor windows. Curve *C* is again the pressure computed from the wind velocity. The ratio of the average pressure difference to the wind pressure is



FIGS. 8, 9, 10. CURVES SHOWING RELATION BETWEEN PRESSURE DIFFERENCE AND WIND VELOCITY FOR 4TH FLOOR WINDOWS

represented by curve *E*, and this is the relation that is the principal objective of this part of the study. Up to a velocity of 11 mph, the pressure difference exceeds the wind pressure, as a result of the temperature effect. Above 11 mph, the pressure difference is less than the wind pressure, and it is interesting to note that curve *E* seems to approach and become constant at about the value of one-half, in the region of high wind velocities. This is just what

would be expected to happen in a building, when the wind pressure becomes so great as to submerge the temperature effect, and where the total leakage area for inflow on the windward faces about equals the total area for outflow on the leeward faces. The inside pressure would have to build up automatically to a value equal to half the wind pressure, so that the head available to force the air out would be equal to the head available to force an equal amount of air in.

Having chosen a value of 20 mph for wind velocity as a basis upon which to determine infiltration in the calculation of heating surface requirements for rooms on exposed sides of the building, curve *E* of Fig. 12 would indicate that the actual pressure difference across the windows is about 0.55 of the pressure computed from this wind velocity, which means that the leakage is about 75 per cent of the amount corresponding to the pressure computed from the wind velocity, since leakage quantity is approximately proportional to the square root of the pressure difference. In other words, the factor to be applied in Part II of the tables of The Guide 1929, to allow for building up of inside pressure should be about 75 per cent in this case.

It will be recalled that the usable maximum wind velocity for sides of the building exposed to the non-prevailing winds was chosen as 13 mph. This is near the value of 15 miles, which is probably the intent of The Guide to suggest for use in connection with the less severe exposures. Referring to Fig. 12, it is seen that for 15 mph, the ratio of actual average pressure difference at the two windows, to the calculated wind pressure, is 0.67, which means that the leakage is about 82 per cent of the amount corresponding to the pressure computed from the wind velocity.

From this study, it appears that the factor of 80 per cent, as applied in Part II of the tables of The Guide 1929, is well chosen, probably erring, if at all, on the side of being too large. However, it is indicated that the practice of allowing an additional 15 per cent in leakage loss for the sides of a building exposed to the prevailing wind may not provide an adequate margin of heating surface on those sides. For example, if 15 mph be taken as the wind velocity upon which to figure infiltration on the less exposed sides, the leakage per foot of sash crack of a double hung weatherstripped wood window is 11.7 cu ft per hour. It has been shown that a reasonable value to be taken for wind velocity from prevailing directions is 20 mph, for which the leakage is 22.9. Thus in the case studied the leakage of windows under the more extreme conditions on the more exposed sides, is nearly twice as much as that on the less exposed sides, instead of only 15 per cent greater. If this is true of infiltration loss, it is also true of transmission loss, although probably in a different ratio. In other words, in the determination of heating surface to take care of the most severe situations, it is probable that there should be a greater differential than 15 per cent in the allowances for infiltration and transmission loss between the windward sides of a building and the sides of lesser exposure.

Distribution of Leakage at Windows at Different Levels

Referring to Fig. 11, it is seen that the pressure difference at the second floor window is considerably greater than it is for the fourth floor, which, as previously explained, is practically accounted for by the temperature difference effect. Converting the pressure difference at 20 mph into corresponding

wind velocity, and then going to THE GUIDE table, it is found that the leakage at the second floor for a double hung weatherstripped wood window is about 17.5 cu ft per hour per foot of sash crack, and that at the fourth floor about 12.8 cu ft which calls for 0.025 sq ft more radiation per foot of crack on the second floor than on the fourth. With eight windows of say 50 ft of sash crack each, including transoms, the total difference in radiation would amount to 10.0 sq ft. Considering windows of poorer construction, and taking frame leakage into account, the total difference in radiation between the floors for rooms at the one end of the building (Fig. 4) might easily run to 40 or 50 sq ft. The difference in heating surface requirements between lower and upper floors is thus seen to be of considerable amount even in the case of three or four story buildings. It would of course be greater still in taller buildings, although it is reduced when there is not free communication between stories by way of open staircases or other means.

In connection with the matter of extra allowance in square feet of radiation to take care of maximum infiltration, it is to be emphasized that this is not a measure of heat loss by infiltration, but is rather an allowance in capacity to meet an extreme or emergency situation. Infiltration goes on to some extent continually, so long as there is either a temperature difference or a wind, but not at the maximum rate. The column of the tables in THE GUIDE 1929 headed *Heat Loss* is likely to be misinterpreted to mean a continual drain from the heating system of the Btu rate per degree temperature difference indicated, which is not the case.

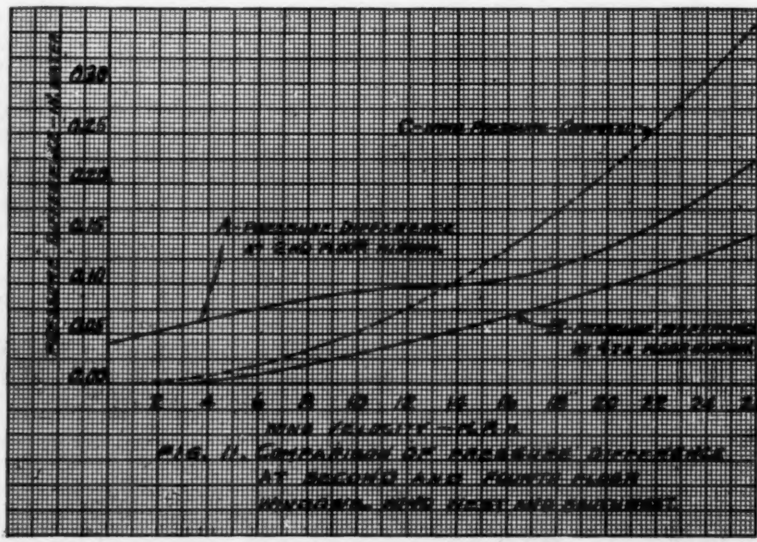


FIG. 11. COMPARISON OF PRESSURE DIFFERENCE AT 2ND AND 4TH FLOOR WINDOWS—WIND W. AND S. W.

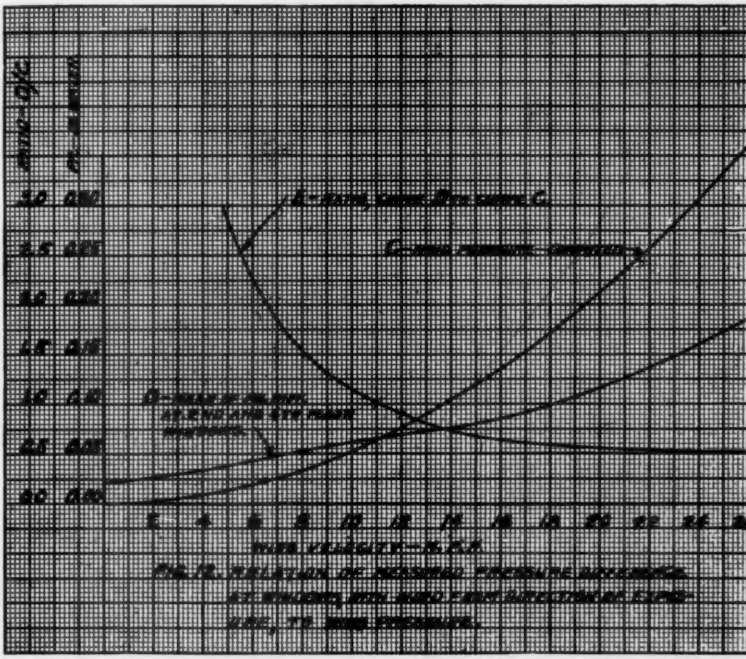


Fig. 12. Relation of Measured Pressure Difference at Windows, with Wind from Direction of Exposure, to Wind Pressure

In conclusion it may be said that the study described and developed in this paper throws some light upon the relation of actual pressure difference across windows to wind pressure, for one particular case. It is indicated that something definite may be derived from a study of this kind. It is further plainly shown that temperature effect, even in buildings of moderate height, may be a factor of sufficient importance to require a different allotment of heating capacity as between lower and upper floors. A study of Weather Bureau records and a general agreement as to the interpretation of data therefrom are highly important in order to better define what shall be considered as the wind velocity upon which to determine window leakage.

DISCUSSION

S. H. Givelber: Is there any relation between the pressure differences and the effect they have on the temperature in the room?

C. G. Segeler: I would like to question something that Professor Emswiler has stated or perhaps point out a source of possible difference of opinion on this particular paper. Professor Emswiler has very carefully selected data for 45 days on both wind direction and velocity. Would the conclusions drawn

be the same if the data had only been collected on such days as might be termed maximum heating days, that is, days on which the given heating system was taxed to the utmost? It would seem to me that the information that is ultimately going to be necessary in deciding such a point is what wind velocity to use in estimating the radiation required for a given building. I feel that it is possible (on this I am not informed; I would like to hear from Professor Emswiler) that the maximum day conditions might be different from those taken over a 45-day period. That is to say, the higher winds might not coincide with the coldest days. That would alter the picture presented.

G. L. LARSON: This paper represents a very important study and I hope that the authors will follow it up with further tests. In laboratory tests, certain pressures are created on the two sides of a wall, or window, and the leakage due to these pressure differences can be very accurately measured. But before a designer can intelligently use the results of these laboratory tests, he must know something about the actual pressure differences that actually exist in practice. This paper throws considerable light on this point, but more research on this subject is necessary. I hope the authors will continue the excellent work that they have started.

The relation between existing winds and outside temperature was mentioned by the last speaker. I think the general practice of THE GUIDE is to assume a certain outside temperature, say —10 or —20 F, depending on the locality, and at the same time to apply a wind velocity of 15 mph. I believe this is wrong because this procedure superimposes two peaks upon each other, and they do not exist together.

I have in mind the records of the heating plant at the University of Wisconsin, where the heating plant is connected to some 93 buildings. The record from this many buildings should give a good average of what is taking place in such a group. The maximum heating load last year occurred on the day when it was four degrees above zero, although it was as much as 25 below several days. At 25 below, or at 20 below, or at 15 below, we ordinarily have practically no wind; the atmosphere is *dead;* but on a day around zero or a little bit above, we have our heaviest winds. So it is questionable whether we should design for —25 F and at the same time add losses due to a wind which is traveling at 15 mph. At least in our section of the country, these peaks do not exist together.

I would like to ask the authors if they tried out different types of tubes to get the static pressure on the outside of a window. I imagine that there might be found considerable difference in the readings depending upon the kind of tube used.

MARGARET INGELS: I believe mention was made that there were gravity exhaust flues in the room. I would like to know if there was any figure taken to find out if the difference in temperature indoors and outdoors caused a variation of air to flow through the gravity exhaust making different pressures build up inside the room.

MR. NICHOLLS: Professor Emswiler speaks of the prevailing winds from the west and southwest. What kind of temperature do you get from that direction as a rule?

PROFESSOR EMSWILER: We usually get our coldest temperatures from the northwest.

Mr. Nicholls: What were they from the south and the southwest?

Professor Emswiler: The average perhaps was 20 F.

Mr. Nicholls: I think you will find that they have a good deal of guessing to do yet when it comes to that. You have to use a factor of wind velocity with your temperature and after you have that worked down to a rational equation, you have to guess what kind of a building you have to heat. Perhaps you will get it some day; I doubt it!

S. R. Lewis: There is in Chicago a 21-story cooperative apartment. Through some mistake the north end of this building, having 21 duplicate apartments, was short of radiation. It was necessary to add radiators. Our observations of the conditions were taken in the second story and the twenty-first, where the complaints originated, but since all of the apartments were exactly alike throughout the 21 stories, I recommended the additional radiation to all of them. When the radiation was shipped and the steamfitters started to work putting them in, just before I left to come to this meeting, the owners of the apartments through the central zone of the building refused to permit the radiators to be added. They said, "We are all right." I suspect it is the infiltration due to the height of the building in the lower part and the exfiltration in the upper part that causes the complaints of shortage of radiation.

L. A. Harding: I think this is a very excellent paper. There were originally two major problems to be solved in reference to infiltration. The first one was to determine the amount of leakage through a crack with a certain pressure difference. That has apparently been satisfactorily solved by these gentlemen and others. The second problem, that of correlating the wind velocity as reported by the Weather Bureau to actual infiltration or pressure difference, remains unsolved. The Research Laboratory is attempting to solve this part of the problem, that is to correlate the wind velocity as reported by the Weather Bureau to actual wind velocity over the surface of building walls. I think you will be very much interested in the paper that our director of research, Mr. Houghten, will be able to present next year along this line. There exists at present, no standard in reference to the distance from the wall surface where the velocity of the air movement is to be measured in order to arrive at surface coefficients to be employed in heat transmission problems relative to building construction.

Mr. Nichols: I think it is going to be extremely difficult to arrive at any thing satisfactory by the method of observation merely of wind velocities. If it is possible to establish some kind of a laboratory so that condensation experiments can be made, under observation of all these conditions, I think a factor can be derived which will have to do with the wind pressure recorded by the Weather Bureau.

Professor Emswiler: In answer to the first question, whether or not there was any relation between pressure difference and temperature in the room, the situation was this: The temperature in the room was held essentially constant by the thermostatic control that operated in connection with the building. If there was more infiltration, that is more cold air coming in, it was compensated for by the thermostatic control. Hence there was, therefore, no necessary relationship as far as that was concerned.

Now, with respect to Mr. Segeler's question, whether the results of such a

study as this ought to be based upon conditions prevailing during a single day of maximum or extreme conditions, or whether it should be based, as we have suggested here, upon a total of several hours during which the wind velocity was up to or above a certain selected level, that is an important question. If you consider the situation from the standpoint of performance, it would seem that you would have to base it upon some such arbitrary factor or time, as we have chosen. If you would want to examine an actual system that is already in operation, then it would be perfectly logical, of course, to observe during that day when the extremes prevailed.

In answer to Professor Larson's question, we have a record of the outside temperature along with wind velocity, but there seemed to be no obvious relation between the two. Of course, with the lower temperature and the same velocity, the inertia or impact effect of the wind is greater. The amount of infiltration, however, is not different with the lower temperature and therefore higher density, but, of course, the weight of air is somewhat different. We did not think this would make a great deal of difference.

The tube used was simply pushed through a hole in the wood strip under the sash, the end of the tube being flush with the outside surface of the board, so that it collected, so to speak, the dynamic pressure that was produced by the wind. It did not of itself produce or result in any localized static pressure as a result of the wind blowing across it, or at same angle to it. It simply, as I see it, collected the static pressure that existed at the exterior of this window due to the dynamic effect of the wind blowing, together with any natural static pressure that prevailed there.

In answer to Miss Ingels' question, we took no measurements of the quantity of air flowing through the gravity flues. We simply took a record of pressure differences as they existed in the two rooms and, these gravity flues would readily account for the displacement of those two pressure difference curves in relation to the wind. They would account for the fact that at zero velocity we found no vacuum on the outside with respect to that inside.

Again, referring to Mr. Nicholl's question, there has already been done a very extensive amount of laboratory work in the determination of the amount of infiltration that will take place through openings or cracks in windows and doors, with certain specified pressure differences applying, and curves and tables present the relation between the quantity of infiltration and the pressure differences. However, I do not see how in any way you could in the laboratory decide upon the question as to what wind velocity you were going to use in applying or making use of the tabular values resulting from the laboratory experiments. It seems to me that must be obtained from statistical data with respect to the observatory records.

MR. NICHOLLS: You have to do some intelligent guessing, you mean?

PROFESSOR EMSWILER: Mr. Lewis and Mr. Harding presented very interesting phases of this same question and I am sure that we all look forward with anticipation to the results that are available when they have been finished. However, the only record which we were interested in was the pressure difference across the window. We are continuing this experiment and possibly may have some more to present at some later time.

No. 851

AIR INFILTRATION THROUGH VARIOUS TYPES OF BRICK WALL CONSTRUCTION[1]

By G. L. LARSON[2] (*MEMBER*), D. W. NELSON[3] (*MEMBER*), *and* C. BRAATZ[4] (*NON-MEMBER*), MADISON, WIS.

The results of cooperative research between the University of Wisconsin and the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

INTRODUCTION

SINCE the fall of 1927, a program of cooperative research concerning air-infiltration through various types of brick wall construction, sponsored by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the University of Wisconsin, has been in progress. The first results of this research appeared as a paper entitled, Effect of Frame Calking and Storm Windows on Infiltration Around and Through Windows, which was presented at the Semi-Annual Meeting of the Society in June, 1928. Preliminary to the report contained herein, a second paper entitled Air Infiltration Through Various Types of Brick Wall Construction, was presented at the Annual Meeting of the Society in January, 1929.

This report is in conclusion to the latter paper and contains results obtained on plain, plastered, and painted brick walls.

DESCRIPTION OF APPARATUS

The test equipment used to determine air leakage or infiltration is shown in Figs. 1 and 2. Briefly, it consists of the following: The pressure chamber, *A*, and the collecting chamber, *B*, between which the wall is placed. The joints between the steel channel frames, in which the walls are constructed, and the chambers *A* and *B*, were made air tight by clamping a rubber gasket between the flange of each chamber and the corresponding flange of the steel frame, in a manner which is shown in Fig. 2. To facilitate setting up a wall for testing, the collecting chamber side of the machine is mounted on a small four-wheel truck which can be moved back about 10 ft from the pressure chamber on a narrow gage track.

Artificial wind pressure is produced by a small motor-driven blower, shown

[1] Final report on Brick Wall Construction, preliminary report of which is published in A. S. H. V. E. TRANSACTIONS, 1929.
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Presented at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1930.

at the extreme left of Fig. 2. The pressure drop through the wall was controlled by means of a damper on the intake of the blower, a relief damper, *D*, and a sliding damper, *E*, located in the connection between the blower and the pressure chamber, at right angles to the flow of air. The pressure drop through the wall, from which the corresponding wind velocity was computed, was measured with an inclined draft gage *F*.

The amount of air which filtered through a wall was measured by interchangeable orifices mounted at the end of orifice box, *C*, and ranging in size from ⅜ in. to 1¾ in. in diameter, accurately machined in accordance with Durley's[5] specifications. A coefficient of 0.6 was applied to all orifices, in conformity with Durley's results. The accuracy of the small size orifices was checked by means of a gas meter. A Wahlen gage, *G*, was used to determine the

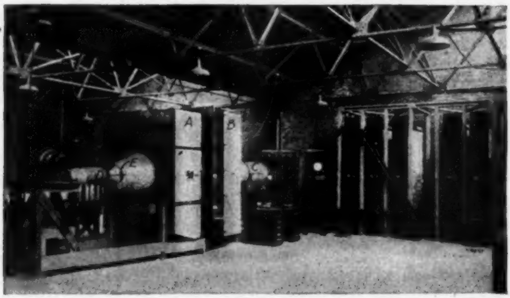

FIG. 1. GENERAL LAYOUT OF TEST EQUIPMENT

pressure difference existing between the orifice box and the atmosphere, into which the measured air was discharged.

GENERAL DESCRIPTION OF WALLS

All walls were built into frames constructed of 15 in., 33 lb steel channels as shown in Fig. 2. The A-shaped frame which appears in the foreground, together with a single roller jack arrangement, attached to the other end of the wall, provides a convenient means of transporting the walls between the test machine and the storage rack, shown in Fig. 1.

Two types of brick were used in the construction of these walls—a hard face brick and a more porous type, commonly known as Chicago clay brick. Water absorption tests were conducted on samples of each type of brick by the National Bureau of Standards, the results of which are tabulated in Table 1. The average dimensions of the hard brick in inches, in the order: Length, width and thickness, were 8⅛, 3¾, 2¹¹⁄₃₂, and for the porous type, 8⅟₁₆, 3⁹⁄₁₆, 2⁵⁄₁₆. The average thickness of the vertical and horizontal mortar joints for all of the walls was approximately ¼ and ½ in. respectively.

Three of the walls were slushed with cement-lime mortar; the others with

[5] On the Measurement of Air Flowing into the Atmosphere Through Circular Orifices in Thin Plates and Under Small Differences of Pressure, by R. J. Durley. Volume 27. *Transactions, A. S. M. E.*

lime mortar. The walls were constructed such as to differentiate between good and poor workmanship. Each of these terms is defined as follows:

Cement-lime mortar: One of cement, one of lime and six of sand by volume, and enough water to make the mixture workable.

Lime mortar: One of lime and three of sand by volume and enough water to make the mixture workable.

Workmanship: Good workmanship is distinguishable from poor workmanship only in the manner of using the mortar. In good workmanship, the spaces between the bricks are completely filled with mortar throughout the thickness of the wall, resulting in a wall which is practically free from voids. In poor workmanship, very little mortar is used between the two outside faces of the wall. The outside appearance of these walls is the same. Fig. 3 is intended to show this difference in workmanship. The poorer wall appears in the foreground.

The walls were constructed by bricklayers from the Service Department of the University of Wisconsin in a manner comparable to actual building construction practice. Prior to their construction, which took place during the month of February, the bricks were stored out-of-doors. In the opinion of the bricklayers, the brick contained the proper amount of moisture, and the addition of more moisture by soaking was deemed unnecessary.

With the exception of Wall No. 2, the mortar joint between the bricks and the steel channel frame was completely calked with a plastic calking compound, on both sides of all walls. Fig. 4 shows the effect of calking one or both sides of this joint for Wall No. 2. The net area of each wall, excepting No. 7, was 50 sq ft.

Procedure

Each of the walls was subjected to wind pressures corresponding to wind velocities ranging from about 5 to 30 mph. With the exception of Wall No. 7, which stood for eight months before testing, the original tests of all plain brick walls were made after an aging period of five months. These tests were repeated at the end of two additional months.

Walls No. 4 and No. 5 were later plastered, on metal lath and furring space, for further testing. After removal of the plaster, Walls No. 4 and No. 5 were used for the determination of the effectiveness of several kinds of paint on infiltration.

Wall No. 6 was also re-tested with an application of plaster directly on the brick, without lath.

Wall No. 7 was used only to determine the effectiveness of plaster, no tests being made on the plain brick wall.

A more detailed description of the conditions under which these tests were conducted relative to plaster composition, nature of paint, aging or drying periods, etc., is to be found in the section on Addition of Plaster and Paint to Plain Brick Walls.

Discussion of Results on Plain Brick Walls

Fig. 5 shows the results of the tests on the plain brick walls. The infiltration in cubic feet per hour per square foot of wall surface is plotted against

FIG. 2. MACHINE OPEN WITH TEST WALL IN PLACE

the pressure drop through the wall in inches of water. Each curve was determined from the points of two series of tests; one set of points from the original tests made five months after construction and the other from the check tests made two months later. The difference in values of the original and check tests was found to be less than one per cent for Walls 3, 4 and 6. The check tests on Wall 2 showed 7.1 per cent less infiltration than did the original tests and the check tests on Wall 5 resulted in 2.6 per cent more infiltration than did the original tests. This seems to show that there is no correlation between infiltration and the aging of the walls between the time of the original and check tests. Further, there seems to be no correlation between the humidity at the time of testing and the variation in test results. This comparison is given in Table 2.

The test results in cubic feet per hour per square foot of wall as shown in Fig. 5 are replotted in Fig. 6 against a uniform scale of velocity in miles per hour instead of against a uniform scale of pressure drop in inches of water.

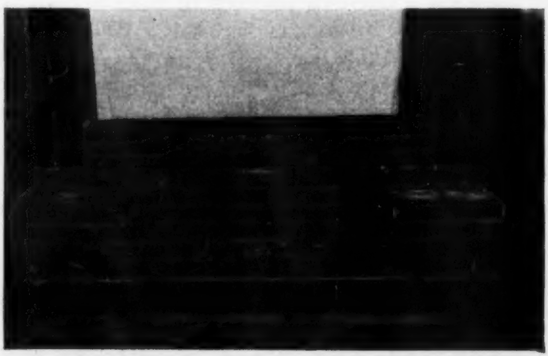

FIG. 3. BRICK WALLS UNDER CONSTRUCTION

When plotted in this way, the results show how rapidly infiltration increases with increase in wind velocity.

Table 3 gives the infiltration through the various plain brick walls for the range of pressures used in the tests and the corresponding wind velocities ranging from 5 mph to 30 mph. This table also shows a summary of the workmanship, mortar and brick for each of the walls.

The results show that Wall 6 is considerably poorer from the infiltration standpoint than the other walls. Wall 6 was constructed of porous brick and lime mortar applied with poor workmanship. By improving the workmanship and using cement-lime mortar rather than lime mortar in Wall 4, the infiltration loss is cut to a little less than 50 per cent. In the case of the hard brick walls, Wall 5, the poorest, allows the passage of 37 per cent as much air as does the poorest wall built of porous brick. The best wall built of hard brick, Wall 2, allows the passage of about 70 per cent as much air as does the poorest wall built of hard brick, Wall 5. The comparison given here between the poorest and best walls for each type of brick is not strictly true, since the poorest of the two hard brick walls had cement-lime mortar as against lime mortar for the poorer porous brick wall.

Table 1. Water Absorption Tests of Bricks

Water Absorption (Total Immersion) as Per Cent of Dry Weight

		1 hr cold	5 hr cold	24 hr cold	48 hr cold	5 hr boiling
Red (Hard) Brick........ 10 specimens..............	Ave.	13.1	13.7	14.3	14.8	16.9
	Max.	14.7	15.3	15.9	16.7	18.8
	Min.	10.0	10.3	10.9	11.3	13.4
White (Porous) Brick.... 10 specimens..............	Ave.	15.0	16.4	17.7	18.2	21.4
	Max.	19.9	21.0	22.0	22.6	24.5
	Min.	11.2	12.8	13.9	14.7	18.3

Penetrability of Brick Expressed as Per Cent of Dry Weight Absorbed in 1 Hour When in ⅛-In. Contact With Water

		Per Cent Absorption			Time to Wet Through	
		End	Face	Flat	End Face	Flat
Red Brick...... 10 specimens....	Ave.	4.7	8.8	13.6	Greater than 1 hr..	33 m
	Max.	5.3	10.1	15.2	Greater than 1 hr..	50 m
	Min.	3.6	6.1	10.4	Greater than 1 hr..	28 m
White Brick.... 10 specimens....	Ave.	6.3	9.6	13.7	Greater than 1 hr..	Greater than 1 hr
	Max.	8.4	13.8	19.1	Greater than 1 hr..	
	Min.	3.8	4.8	10.4	Greater than 1 hr..	40 m

Had a wall been built of hard brick, lime mortar and poor workmanship, its probable leakage would have been 4.59 cu ft per hour per square foot at 15 mph. This figure is arrived at by adding to the leakage through Wall 5 which was built of hard brick, with cement-lime mortar and poor workmanship, the difference between the leakage of lime mortar and cement-lime mortar as

applied to hard brick Walls 2 and 3 (3.85 + 0.74 = 4.59). The true comparison then between the poorest and best hard brick walls would be that the best hard brick wall would have a leakage value a little less than 60 per cent of that of the poorest hard brick wall. The corresponding comparison for porous brick walls was slightly less than 50 per cent. At 15 mph, the difference between the best and poorest of the hard brick walls would be 4.59—2.71 = 1.88 cu ft per hour per square foot of wall, and for the porous brick walls would be 10.35 — 5.05 = 5.30 cu ft per hour per square foot of wall.

This comparison shows that there is a greater variation in infiltration between the best and the poorest walls built of porous brick than between those

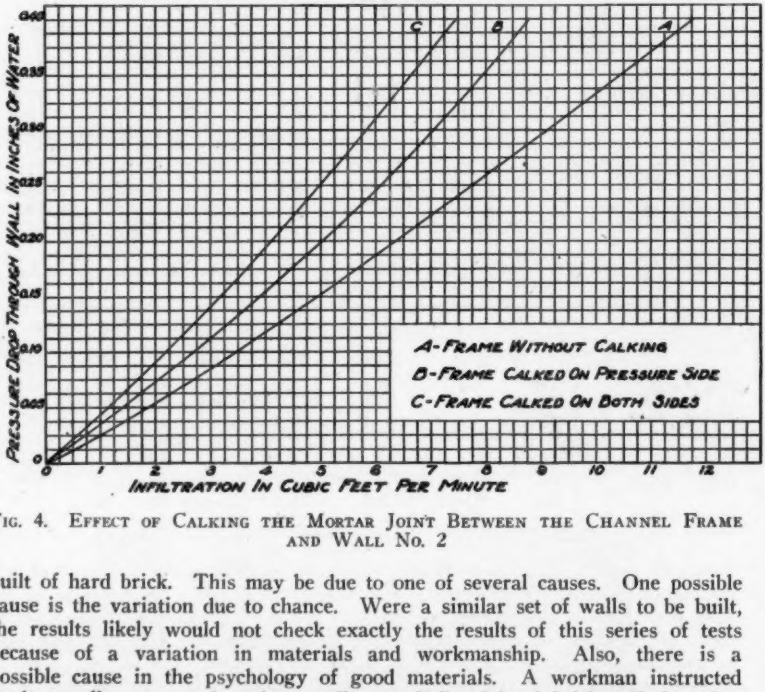

FIG. 4. EFFECT OF CALKING THE MORTAR JOINT BETWEEN THE CHANNEL FRAME AND WALL NO. 2

built of hard brick. This may be due to one of several causes. One possible cause is the variation due to chance. Were a similar set of walls to be built, the results likely would not check exactly the results of this series of tests because of a variation in materials and workmanship. Also, there is a possible cause in the psychology of good materials. A workman instructed to do equally poor work on two walls, one built of hard brick and the other of porous brick, might unconsciously do better work with the better material, the hard brick. It also seems logical to believe that two walls of good workmanship are more likely to be on a comparable basis from a workmanship standpoint than would two walls of poor workmanship, inasmuch as a completely slushed wall is more easily duplicated.

Since poor workmanship consists mainly in leaving voids between the bricks in the interior of the wall, and since the porous brick is likely to be much less uniform in density, it may be that poor workmanship allows passageways

for air through short distances from the face of the porous brick to the voids, and then out on the other face of the wall through a short distance of brick. This explanation would require that the hard brick wall passes most of the total infiltration through the mortar joints.

Another probable cause for this greater variation between the best and poorest of porous brick walls as compared to the best and poorest of hard brick walls is in the effect of the porosity of the brick on the proper setting of the mortar. It is likely that the porous brick draws the water from the

TABLE 2. COMPARISON OF HUMIDITY AND TEST RESULTS VARIATIONS

Wall No.	Av. Humidity Original Tests	Av. Humidity Check Tests	Change in Humidity Expressed in Per Cent	Variation in Test Results Per Cent
2	74.3	40.5	— 45.4	— 7.07
3	55.9	63.4	+ 13.4	— 0.27
4	77.2	58.8	— 23.8	— 0.14
5	65.4	31.7	— 51.6	+ 2.62
6	67.1	36.6	— 45.5	— 0.24

TABLE 3. INFILTRATION IN CUBIC FEET PER HOUR PER SQUARE FOOT OF PLAIN WALL

Drop in Pressure Inches of Water	Wind Vel. mph	Wall No. 2	Wall No. 3	Wall No. 4	Wall No. 5	Wall No. 6
0.012	5	0.34	0.46	0.71	0.51	1.60
0.048	10	1.30	1.64	2.36	1.83	5.30
0.108	15	2.71	3.45	5.05	3.85	10.35
0.192	20	4.59	5.76	8.31	6.34	16.28
0.300	25	6.85	8.38	12.03	9.22	23.05
0.431	30	9.31	11.30	16.00	12.40	30.80

Wall No.	Kind of Workmanship	Kind of Mortar	Kind of Brick
2	Good	Cement-lime	Hard
3	Good	Lime	Hard
4	Good	Cement-lime	Porous
5	Poor	Cement-lime	Hard
6	Poor	Lime	Porous

mortar before it has time to set and consequently causes an opening of pores and a shrinking away of the mortar from the brick surfaces. The bricks, both hard and porous, were considered by the mason to contain the proper amount of moisture for the proper laying up in the walls.

Table 1 shows the water absorption characteristics of the two kinds of brick used. It seems that the absorption test giving the best indication of the drying out effect of the bricks on the mortar would be the total immersion test in cold water for 24 hours, since the bricks are about five-sixths immersed in the mortar and 24 hours is near to the time required for proper setting of the mortar. For total immersion for 24 hours in cold water, a comparison of the figures in Table 1 shows that the hard bricks absorb 81

per cent as much water as the porous bricks. In the absorption tests, the bricks are dry to start with; in building the test walls, the bricks come in contact with mortar when they are somewhere between the dry and saturated condition. Also one face is exposed to the air. Under these altered conditions, it is not known whether the ratio of absorption for the two types of bricks would remain the same as under the absorption test conditions. The tests on air infiltration seem to indicate a greater difference in leakage through mortar applied to hard and porous bricks than the absorption tests would indicate.

The hard brick wall, with lime mortar and poor workmanship synthesized from Wall 5 and a comparison of Walls 2 and 3 would have a probable

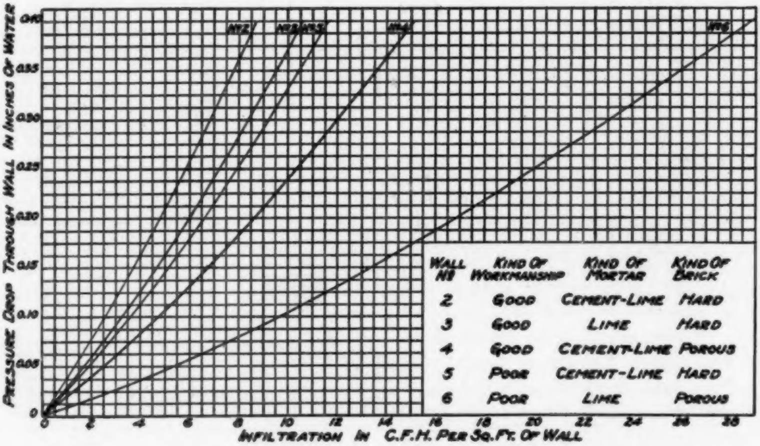

FIG. 5. RESULTS OF INFILTRATION TESTS ON THE 13-INCH PLAIN BRICK WALLS

leakage of 4.59 cu ft per hour per square foot at 15 mph. This would be a wall built to the same specifications as Wall 6, except for the difference in brick. Wall 6 had a leakage of 10.35 cu ft per hour per square foot at 15 mph. The poorest hard brick wall would then have a leakage of 44 per cent as great as that through the poorest porous brick wall.

The substitution of cement-lime mortar for lime mortar in this poorest hard brick wall would reduce the leakage by 0.74 cu ft per hour per square foot of wall at 15 mph, or a saving of 16 per cent. The difference in leakage of Walls 5 and 2 gives the comparison of good and poor workmanship for hard brick walls. The saving in using the better mortar is 1.14 cu ft per hour per square foot, or a saving of about 25 per cent. The total reduction in infiltration by using the cement-lime mortar applied with good workmanship in place of lime mortar applied with poor workmanship is 1.88 cu ft per hour per square foot for hard brick walls, or a reduction of 41 per cent.

The same comparison of mortar and workmanship cannot be made separately for porous brick walls, since only two walls were tested. However, the best

porous brick wall on which cement-lime mortar was applied with good workmanship has a leakage of 5.30 cu ft per hour per square foot less than does the poorest porous brick wall, which uses lime mortar applied with poor workmanship, or the reduction is 51 per cent. For hard brick, this reduction was 41 per cent. This seems to indicate that it is more important to use the best mortar and workmanship on porous brick walls than on hard brick walls. This is indicated both by the greater saving in cubic feet of infiltration and by the percentage saving. In actual construction, the item of cost would also enter in choosing material and workmanship. The hard brick is more expensive and the added cost of cement-lime mortar over lime mortar, and good workmanship over poor, would result in a smaller percentage increase in the wall cost than in the case of porous brick walls.

A hard brick wall with lime mortar applied with poor workmanship, it was found, would have a leakage of 4.59 cu ft. The porous brick wall with cement-lime mortar and good workmanship has a leakage of 5.05 cu ft. Then the poorest hard brick wall has a leakage 91 per cent as large as that of the best porous brick wall.

Addition of Plaster and Paint to Plain Brick Walls

With the completion of the original and check tests on the plain brick walls, the results of which are shown in Table 3, plaster and paint were applied to some of the walls. The walls used for these additional tests were numbers 4, 5 and 6, and were selected because of their relatively high leakage as plain walls. The same wall was used for both plastering and painting in the case of Walls 4 and 5. The work was done by plasterers and painters from the Service Department of the University of Wisconsin.

The first step taken with Walls 4 and 5 was the application of metal lath and plaster with a furring space. The thickness of the plaster was ¾ in. and the furring space was about the same. The plaster used was a gypsum plaster applied in three coats. The scratch coat consisted of two parts of sand to one part of plaster with hair. The brown coat was proportioned two and one-half parts of sand to one part of plaster with hair. The sand finish coat was made of equal parts of sand and plaster without hair. On Wall 6, three coats of gypsum plaster were applied directly to the brick. The total thickness of the plaster and the composition of the coats were the same as for Walls 4 and 5, except that the brown coat was proportioned three parts of sand to one part of plaster with hair. The walls were allowed to stand three weeks or longer after plastering before testing. The joint between the steel frame and the plaster was sealed with roofing cement.

After the tests of Walls 4 and 5 with the plaster applied were completed, the walls were stripped of the furring, lath and plaster, so that they were in their original plain condition except for the nail holes remaining in the mortar joints from the nailing of furring strips. The walls were re-tested for air infiltration and paint was then applied. All painting was done with a brush.

Three coats of exterior white lead and oil paint were applied to Wall 4 directly on the bricks. This paint was mixed in the proportions of 100 lb of white lead paste to 4 gal of linseed oil and ½ gal of turpentine. The wall was tested after the application of each coat. The first and second coats were

applied with just ordinary care with no attempt to fill all crevices. The wall appeared well painted after the application of the first two coats. The third coat was put on with special effort to seal all pores and crevices. This coat was applied with much more than ordinary care. A drying period of one week was allowed after each coat before testing.

One heavy coat of cold water paint was applied directly to the bricks of Wall 5 upon removal of the furring strips, metal lath and plaster. The wall was tested after a drying period of three weeks. After the wall was tested, this coat was washed off as completely as possible with a stiff brush and water; first by hard scrubbing to loosen the paint, and then by flushing off with water.

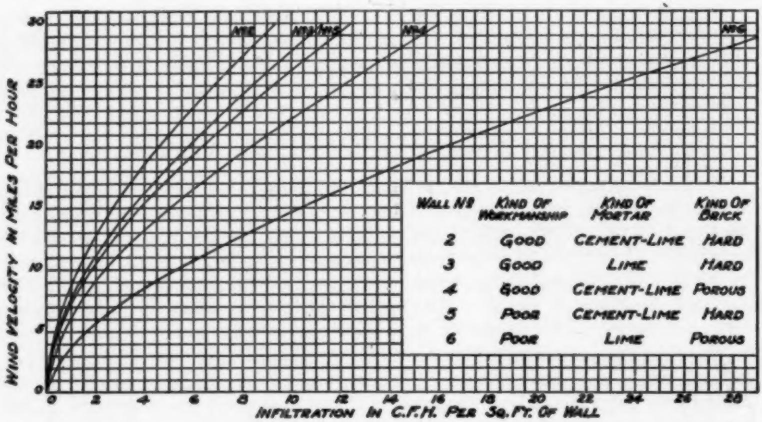

FIG. 6. CHART SHOWING HOW INFILTRATION INCREASES WITH INCREASED WIND VELOCITY THROUGH 13-INCH PLAIN BRICK WALLS

An effort was made to remove it from the mortar joints. The wall was then tested again after standing one month.

Wall 5 was then given two coats of a prepared linseed oil paint containing barium sulphate and zinc sulphide mixed with turpentine for interior use. A test was made after the application of each coat. The first coat dried for two and one-half weeks before testing; the second, five days.

No tests were made of paint applied to plaster.

The results of the plaster and paint tests are shown in Table 4 for Wall 4, Table 5 for Wall 5 and Table 6 for Wall 6. The results are shown graphically in Fig. 7 for Wall 4, Fig. 8 for Wall 5, and in Fig. 9 for Wall 6. The plain wall results before plaster was applied and after plaster was removed have been included in these tables and on the curve sheets for comparison. The one exception to this was in the case of Wall 4, where the plain wall results before plastering were left out, since the curve was almost identical to *C*, the curve for one coat of white lead and oil.

The results show that in each case plastering reduced the infiltration to a

very low value. At 15 mph, plastering reduced the leakage through Wall 4 by 94 per cent, through Wall 5 by 96 per cent, and through Wall 6 by 98 per cent. It is to be expected that plaster applied directly to the brick would be more effective in preventing infiltration than plaster applied on lath with a furring space, since the furring space acts as a distributing chamber for

TABLE 4. RESULTS OF TESTS ON WALL 4

Wind Vel. mph	Drop in Pressure Inches of Water	Plain Wall	Furring Metal Lath Plaster	Plaster Removed	1 Coat White Lead and Oil	2 Coats White Lead and Oil	3 Coats White Lead and Oil
5	0.012	0.71	0.05	0.65	0.61	0.61	0.52
10	0.048	2.36	0.14	2.52	2.38	2.31	1.87
15	0.108	5.05	0.28	5.22	4.97	4.76	3.75
20	0.192	8.31	0.47	8.64	8.31	7.78	6.08
25	0.300	12.03	0.68	12.55	12.07	11.27	8.69
30	0.431	16.00	0.92	16.87	16.03	15.18	11.62

TABLE 5. RESULTS OF TESTS ON WALL 5

Wind Vel. mph	Drop in Pressure inches of Water	Plain Before Plastering	Furring Metal Lath and Plaster	Plain After Plaster was Removed	1 Coat Cold Water Paint	Cold Water Paint Washed Off	1 Coat White Oil Paint	2 Coats White Oil Paint
5	0.012	0.51	0.02	0.65	0.27	0.27	0.27	0.27
10	0.048	1.83	0.07	2.30	1.03	1.03	1.03	1.03
15	0.108	3.85	0.15	4.61	2.22	2.17	2.13	2.03
20	0.192	6.34	0.25	7.45	3.77	3.37	3.27	3.08
25	0.300	9.22	0.34	10.87	5.58	4.54	4.40	4.12
30	0.431	12.40	0.42	14.86	7.67	5.77	5.61	5.19

TABLE 6. RESULTS OF TESTS ON WALL 6

Wind Vel. mph	Drop in Pressure Inches of Water	Plain Wall	Plastered Directly on Brick
5	0.012	1.60	0.03
10	0.048	5.30	0.09
15	0.108	10.35	0.18
20	0.192	16.28	0.29
25	0.300	23.05	0.40
30	0.431	30.80	0.52

air coming through passageways in the brick and mortar. Plaster applied directly, it seems, would tend to close the passageways in the brick and mortar. The percentage figures given above seem to indicate that this is true, since the percentage reduction was the greatest (98 per cent) for Wall 6 which had plaster applied directly. Not too much reliance can be placed on the difference in these percentage reductions because of the small quantity of air being measured in the tests of plastered walls. For example, at 15 mph for Wall 5, a reduction in leakage of 0.03 cu ft per hour per square foot of wall would change the percentage reduction by one per cent. It would seem

that the most difficult condition in which to secure a large reduction in leakage would be by plastering on the wall having the greatest leakage as a plain wall. Wall 6 had considerably the highest leakage as a plain wall, which seems to indicate that the greater per cent reduction for plaster applied directly to the brick is real. In service, however, a furred wall would probably have less tendency to crack from any settling occurring in the brick wall and would, therefore, better maintain its resistance to air leakage.

The application of one coat of interior cold water paint reduced the leakage at 15 mph of Wall 5 by 52 per cent. The wall face was completely covered by this coat and no second coat was applied. The reduction in leakage on

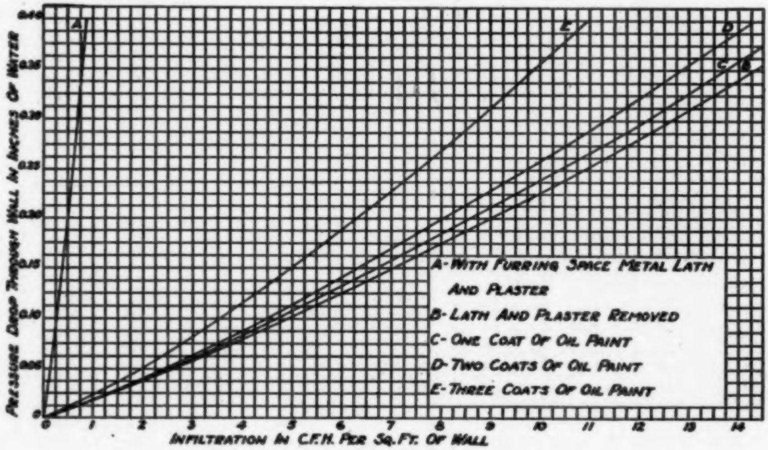

FIG. 7. INFILTRATION THROUGH WALL NO. 4 UNDER VARIOUS CONDITIONS

painting with cold water paint is considered to be due to the clogging of pores and crevices in the brick and more especially in the mortar joints by the paint which was applied as a thick paste. The surface appearance was dull and chalky and gave no indication of effecting a considerable reduction in leakage. Upon testing after the removal of all cold water paint from the surface by scrubbing with a brush and water, the leakage was found to decrease slightly over the leakage secured when the cold water paint was applied. This further bears out the belief that the leakage was reduced by filling openings below the surface. The scrubbing action with water apparently carried additional paint material into pores and crevices.

Wall 5, then in the condition of a plain wall with cold water paint washed off, was treated to two coats of a white oil paint. It would seem that the cold water paint, although filling the crevices, would still be porous and that the oil from the oil paint would bind this porous material together and further reduce the leakage considerably. The first coat gave a reduction in leakage at 15 mph of only 2 per cent and the second coat an additional 4.5 per cent, or

a total reduction of 6.5 per cent. The application of two coats to the hard brick seemed to give a smooth and finished surface. The surface was less perfect on the mortar and it would seem that the oil paint was incapable of stopping the leakage through pores and crevices that the cold water paint had not already filled and stopped completely. White lead and oil paint applied to the bare brick surface of Wall 4 after the furring strips, metal lath and plaster were removed gave a reduction measured at 15 mph of 5 per cent for one coat and 9 per cent for two coats. The brick of this wall was a porous brick and it was observable that two coats did not cover all the imperfections in the brick, although the appearance of the surface was excellent from a

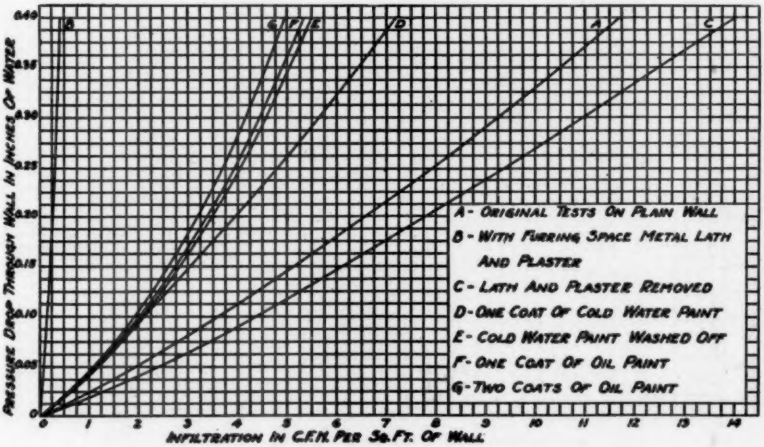

FIG. 8. INFILTRATION THROUGH WALL NO. 5 UNDER VARIOUS CONDITIONS

distance. An additional coat applied with extreme care in an attempt to cover all imperfections resulted in an additional 19 per cent reduction or a total reduction of 28 per cent for the three coats.

The nail holes made by the nails holding the furring strips in place increased the leakage considerably more on the poor workmanship wall, No. 5, than on the good workmanship wall, No. 4. The test of Wall 4 after the furring strips, metal lath and plaster were removed showed an increase in leakage over that in the original tests attributable to the nail holes of 3.5 per cent measured at 15 mph. The corresponding increase for Wall 5 was 20 per cent. The type of mortar was the same for the two walls. In the case of Wall 4, the joints were completely slushed, whereas in the case of Wall 5, they were not and the nail holes penetrated into the void space in the interior of the wall.

IMPORTANCE OF SEALING PLASTER

The tests show that plaster is very effective in reducing leakage to a negligible value. The condition of the plastered walls tested was most favor-

able to low leakage. No cracks existed in the plaster, and the perimeter against the steel frame channel members was completely sealed. In ordinary building construction, it was doubtful if the full efficiency of the plaster in stopping leakage is ever realized. There is opportunity in the case of a furred wall for the air to travel in the furring space to edges of the plaster sheet and enter the room at the edge of the wood trim such as baseboards, and also to get into the floor construction and enter the room through the flooring joints or through openings made in it such as are made for heating pipes. To obtain the full efficiency of the plaster in stopping air leakage, the plaster sheet should be sealed at all edges. At the baseboard, this would mean the

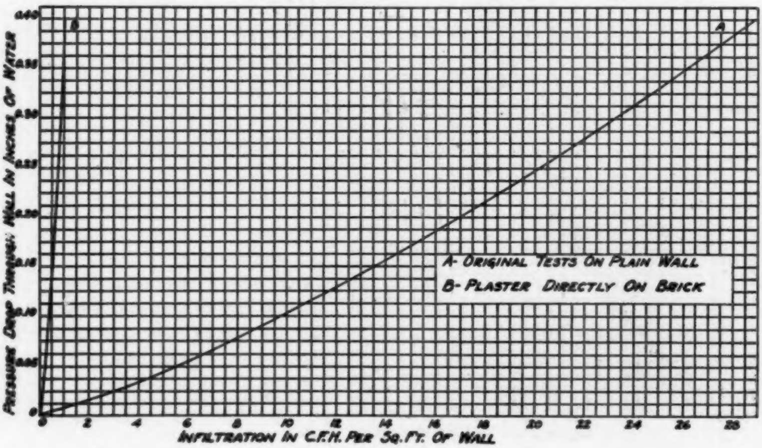

FIG. 9. INFILTRATION THROUGH WALL NO. 6—PLAIN AND PLASTERED

running of the plaster coats down to the rough floor and also filling the space between the plaster sheet and the brick wall surface with plaster. Any cracks or imperfections due to uneven settling of the building would tend to destroy the effectiveness of the plaster in stopping air infiltration.

To study the effect of imperfect sealing at the baseboard, Wall *7* was plastered and equipped with an 8-in. baseboard. The wall was built of porous brick, lime mortar and poor workmanship and was equipped with a window opening. Gypsum plaster in ¾-in. thickness was applied on metal lath in three coats, and a furring space of about the same thickness was provided. The plaster coats were stopped irregularly within an inch or two of the bottom channel, which corresponds to the floor level in building construction. The baseboard was then applied in the usual manner, except it was screwed to the grounds instead of nailed. The wall was allowed to dry one week before testing. Fig. 10 shows Wall *7* after plastering.

Three tests were made of the plastered wall equipped with the baseboard. In test run *A*, the baseboard was fitted as closely as could be against the sand

finished plaster. In this case, the end and bottom joints between the baseboard and wall were completely sealed. The baseboard was routed out or relieved in the center so that the bearing on the wall was over a width of 1¼ in. at the top and bottom. In test run *A*, the fit to the wall was such

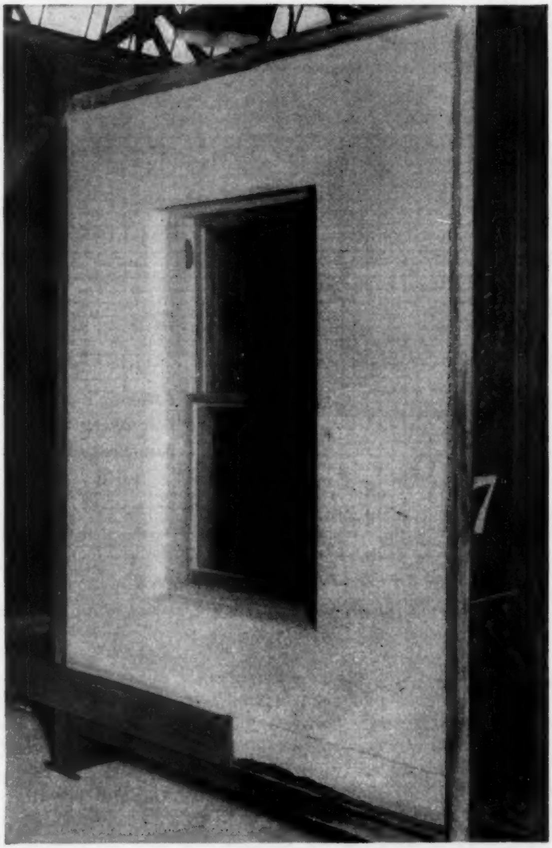

FIG. 10. PLASTER AND BASEBOARD CONSTRUCTION ON WALL No. 7

that at no place could a 1/32-in. gage be slipped between the baseboard and the plaster.

It is estimated that the average clearance was almost 1/32 in., allowing for the roughness of the sand-finished plaster. In test run *B*, the baseboard was completely sealed on the top as well as on the bottom and ends. This cor-

responds then fairly closely to a test of the wall when completely plastered and sealed at the joint against the steel frame members. In test run *C*, the baseboard was removed completely and the furring space was open to the collecting chamber or room side of the testing machine through the inch or two of irregular space at the bottom of the plaster sheet. This run then corresponds to a test of the plain wall not plastered, since all parts of the brick and mortar surface have access to the furring space.

Fig. 11 shows the results of these three test runs in graphical form. The results are given in total infiltration through the wall in cubic feet per minute.

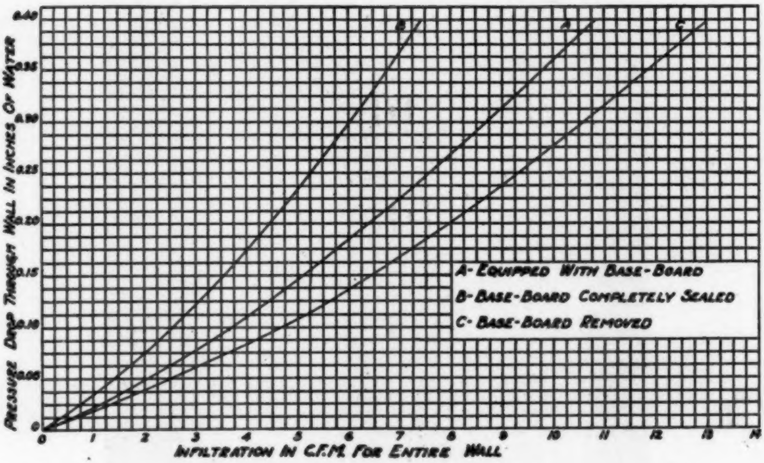

FIG. 11. INFILTRATION THROUGH WALL NO. 7 UNDER VARIOUS CONDITIONS

The difference between the values on Curves *C* and *B* gives the reduction in leakage that is due to plastering completely, allowing no leakage around the baseboard. The difference between the values on curves *C* and *A* shows the reduction in leakage occurring when the plaster is not perfectly sealed at the bottom and a baseboard is applied. The reduction in leakage for plastering completely (*C-B*) is 2.27 cfm at 15 mph for the entire wall. The reduction in leakage over that of the plain brick wall for the condition where the plaster is not perfectly sealed at the baseboard is *C* — A, equal to 1.06 cfm at 15 mph for the entire wall. The effectiveness of the imperfectly sealed plaster and baseboard is then measured by the ratio of $\frac{C-A}{C-B}$, which is about 47 per cent. Then, the plaster and baseboard as built reduced the leakage roughly one-half as much as would the best plaster coat. This seems to bear out the importance of sealing the edges of the plaster sheet as at the baseboard, if the value of plaster in stopping air infiltration is to be realized in building construction.

To study the relative importance of infiltration through brick walls as compared to other factors involved in heat loss calculations, the following example has been worked out. The comparison was made on a ten-story office building omitting the basement. The total boiler load figured 12,200 sq ft of direct radiation for a temperature difference of 70 deg and an emissivity of 240. The wall area, exclusive of window and door openings, was 37,860 sq ft and in this example was considered as all 13-in. brick wall with furring space, metal lath and ¾-in. plaster. The infiltration factors used on the windows and doors averaged about 110 Btu per hour per foot of crack.

An effectiveness of plaster of 50 per cent was used in accordance with the baseboard tests on Wall 7 included in this paper. For Wall 4, this would be an infiltration value of 8.46 cu ft per square foot per hour, or a heat loss factor of 10.7 Btu per square foot per hour for 70 deg temperature difference at 15 mph. For Wall 5. the corresponding values are 2.00 cu ft and 2.5 Btu. Assuming one-half of the wall area exposed to wind at one time, the calculated radiation to overcome infiltration through the brick on Wall 4 would be 926 sq ft or 7.6 per cent of the total calculated radiation, and for Wall 5 would be 220 sq ft, or 1.8 per cent of the total calculated radiation.

The infiltration heat loss through the brick, based on the results of tests on Wall 4, was 25 per cent of the infiltration through windows and doors, and the corresponding figure for Wall 5 is 6 per cent.

The comparisons probably represent conditions in a building where care has not been taken in sealing the plaster sheets at the baseboards and other wood trim. For a case where the sealing of the plaster is perfect, the amount of infiltration through the walls would be greatly reduced.

Based on the test results for Wall 4, the heat loss due to the infiltration through the walls would be 2.7 per cent of the infiltration loss through the windows and doors and the corresponding loss using the results of Wall 5 would be 0.5 per cent. Expressed in terms of the total heat losses for the building, the heat loss due to infiltration through the walls would be 0.8 per cent for the type of construction corresponding to Wall 4 and 0.14 per cent for Wall 5.

This indicates the importance of reducing infiltration for brick walls furred and plastered by the proper sealing of the plaster sheet at the baseboard and other wood trim.

CONCLUSIONS

Plain brick walls vary greatly with respect to air infiltration. Of the three factors—brick, mortar and workmanship—workmanship seems to be of most importance, the composition of the mortar as to cement and lime content next in importance, and the brick the least important. The infiltration for the 13-in. plain brick walls tested ranged from 2.71 to 10.35 cu ft per hour per square foot of walls.

Gypsum plaster, when properly applied, stops almost all infiltration. The results show that plastering stopped about 96 per cent of the leakage of the plain brick wall. Plastering directly on brick seemed to be slightly better than on metal lath with a furring space. The difference is small and the saving due to plastering directly on the brick would be negligible as com-

pared to the saving in heat transmission effected by the furring space in the other construction.

The effectiveness of plaster in stopping infiltration in actual building construction is probably much less than that found in these tests. Cracking of the plaster, imperfect sealing of the plaster sheet at the wood trim, such as at the baseboard, would decrease the effectiveness greatly. A test made of a baseboard equipped wall, with a furring space, showed an effectiveness of only 50 per cent of that obtained with a perfectly sealed plaster wall.

Two ordinary coats of a linseed oil paint applied directly to the surface of a porous brick wall reduced the leakage by 9 per cent. A third coat applied with extreme care made the total reduction in infiltration 28 per cent.

One heavy coat of cold water paint applied directly to the surface of a hard brick wall reduced the leakage by 50 per cent.

A study made on a typical 10-story office building, considering all walls to be 13-in. brick with furring space, metal lath and plaster, indicates that the infiltration heat loss through the brick would range from 6 to 25 per cent of the infiltration heat loss through the windows and doors where the plaster is not properly sealed. For perfect sealing of the plaster, these percentages would be reduced to 0.5 to 2.7 respectively. This would indicate that the infiltration loss through brick walls with properly sealed plaster at the baseboard and other wood trim is negligible in making calculations for heat loss.

DISCUSSION

EDWIN C. EVANS (WRITTEN): This entire matter of building wall leakage is so vitally important to all human beings, affecting as it does the majority of building trades and manufacturers of building materials, I hope the Committee on Research will request a continuation of this investigation by Professor Larson.

Porosity is a natural enemy of practically all of the engineering professions and Professor Larson has shown how far-reaching is the porosity of 13-in. brick walls and he, no doubt, intends to further investigate a simple means or method of eliminating it, perhaps by applying common lead or zinc paint pigment, with special attention paid to the application of the first and second coats; that is, the first coat should be thinned with turpentine and applied without driers to assure penetration of the paint pigment to some given depth below the surface. The second coat should be applied with less turpentine, a little linseed oil and very little drier, while the third coat should be of the proper consistency with the addition of the correct amount of turpentine, linseed oil, and drier, well-known to the trade. This treatment is equally adaptable to concrete and has frequently been used with success to prevent loss with pressures below ½ lb per sq in.

Professor Larson describes his method of application and kind of paint together with the proportions of oil and turpentine, and in Table 4, results are shown that would not justify the use of oil and lead paint to kill porosity. However, if a further study of the application of oil and lead paint can be made, it will be found that painting in this manner will be far cheaper than plaster with even better results than are shown in Fig. 8. In the case of the

water paint, porosity of the water paint itself, is very readily detected (see Table 5) but water paint could have been made to serve its purpose with better results, had the first coat been sufficiently diluted to permit it to flow into the outer surface of the wall.

In the foregoing, I have assumed that all the walls were treated with both paint and plaster on the pressure side without the fan in operation. This point should be brought out, as the pigment or plaster would have the backing of the wall which, in the case of some pigment materials, could readily be judged as ineffective for the purpose intended.

Applied to brick pent house walls, Professor Larson's paper should be very closely studied by architects and heating and ventilating engineers, some of whom, I regret to say, continue to use the pent house room as an exhaust chamber with a housed fan operating as an exhauster at a pressure of ½ in. of water and higher which, due to wall leakage, makes impossible either a reasonable close heat and humidity control or a clean air system. Such pent house designs should be discarded, especially so when studying the tables of air leakage shown by Professor Larson even though he has used wind pressures only up to and including 30 mph, whereas many of us have frequently had to work with pressures equal to wind velocities as high as 85 mph.

Professor Larson has shown the effectiveness of furred lath and plaster wall protection against porosity but only, of course, of gypsum plaster and these values, very likely, will not hold if soft acoustic plasters are used.

Where both the architect and engineer are called upon to contend with air leakage through the wall itself and acoustics within the room or rooms themselves, the reason for further research is apparent and to date, we have only the work of Professor Larson's laboratory to thank for the best known progress. Especially valuable are the results shown in Fig. 11 as regards sealing the baseboard, also his description for the benefit of the architect and trade of running the plaster down to the floor.

The brick mason must use water, and the Bureau of Standards table, shown as Table 1, gives plenty of room for thought on the part of the heating and ventilating engineer and especially the heating contractor who, in an 8 room residence built of 13-in. brick walls, must contend with excess water contained in the bricks and mortar joints. Then the plasterer comes along with a considerable amount of additional water and, while this point is not new to any heating engineer or heating contractor, one must at least be partially surprised when reading the figures on the amount of water that such a residence starts out with. Evaporation of this water, due to outside wind pressure and radiant heat from the sun, will always be an unknown quantity but after a fashion, this job is finally licked with salamanders or some kind of temporary heat.

How often do we hear complaints against a heating system in a new building which proves to be entirely satisfactory after the building is dried out to normal water content. The heating engineer should bear this point in mind and protect the heating contractor when this matter of building water is overlooked in the complaint of the architect or owner.

I want to commend Professor Larson and his co-workers on this very valuable and interesting paper which shows the results of a tremendous amount of work, time and study, and I am sure that all who have studied the charts,

tables and descriptions will join me in this commendation and hope for a continuation of this research.

Were the test walls weighed when first built and at the time they were under test? If so, what was the loss of water weight and what was the additional loss of weight at the end of the test?

Were the paint and plaster applied to the front or pressure side of the wall?

What were the dry-bulb and wet-bulb temperatures of the air entering the fan from the room?

Will tests be run with higher wind pressures up to 60 mph, or say—1¾ in. of water?

L. B. LENT (WRITTEN): If any discussion of this excellent paper is warranted, it may be worth while to call attention to some of the points which might, in my opinion, be worthy of special emphasis.

I think we should bear in mind that a total of only seven walls were built and tested and that each of these seven walls were purposely built to represent a certain set of conditions. It is important to remember that all brick are not alike, but there are many grades varying over a considerable range in physical properties, and there are also many grades of brickwork.

It is quite impossible to construct brick masonry so that what might be called average conditions are represented in a single wall. Average values can only be obtained by averaging the results of the tests of many walls.

In a discussion of a former paper on this subject presented at the annual meeting in Chicago, I offered some information which came from investigations on the same subject made in Germany, and in which it was shown that the amount of air passing through a mortared wall is far in excess of that which one calculates from the data on a single brick; the ratio being approximately 380 to 1. In other words, the amount of air which could be actually forced through a single brick, or through all of the bricks constituting a wall construction, would probably be very much less than that which passes through the mortar joints. Hence, the physical properties of the brick which constitute the wall may be assumed to be of minor importance.

In this paper the average absorption of the hard brick, measured by a 48-hour immersion, was 14.8 per cent and that of the more porous brick measured by the same method, 18.2 per cent. The percentage of absorption for the hard brick is relatively high as compared with that for other hard bricks from other parts of the country.

I am of the opinion that, in any case, absorption percentage has little or no significance in the amount of air which may pass through the bricks, but that it may influence the adhesion of bricks and mortar and, therefore, the tightness of the mortar joint.

Without discussing the results stated in the paper, it would appear, as might be expected, that joint filling and adhesion of mortar to bricks are the important factors affecting the infiltration of air through any brick wall.

Since only one wall of each kind was tested, the results reported may well be considered as indicative rather than conclusive, for reasons already stated. It is also sometimes dangerous to deduce other results than those observed by mathematical calculations, for the actual phenomena may be different from those imagined.

To me, an important conclusion given in the paper is that the tests show that plaster is very effective in reducing leakage to a negligible value. This is important because it would seem to be always desirable to plaster the surface of any brick wall in any structure where heating costs are of any considerable amount. The cost of a plaster coating should much more than offset the cost of any heating equipment plus the annual fuel bill. One can hardly imagine a case where a plaster coating would not be more than justified.

The experience of other research workers in the field of masonry wall strengths, sound resistance, and heat transmission, indicates that much care should be exercised in using the results of tests from *only one sample* of wall construction.

If the results of this investigation are published in THE GUIDE for the information of designers, I think a word of caution should be inserted to the effect that they represent the result of only a single test and, therefore, are not conclusive.

JUDSON VOGDES: We of the Brick Association have been conducting a lot of tests on subjects very closely allied with the infiltration of air through brick walls. This is covered mainly in our investigation of leakage of moisture through walls. We have found that it is very hard to duplicate conditions by taking the same materials and the same workmen and building two walls. There are many factors which enter into the leakage, and I believe it also applies to leakage of air through these walls. As far as the passing of air or water through the individual bricks is concerned, I do not believe that it is a very important consideration, but that workmanship, kind of mortar, are of great importance. We are constantly trying to educate the masons and contractors to build better walls and I believe when they do that there will be less infiltration of air through those walls.

L. A. HARDING: There is one sentence in Mr. Lent's comments on Professor Larson's paper which I question and that is to the effect that one might infer from Professor Larson's paper that all brick walls in buildings which are to be heated should be plastered. I doubt very much whether that is so, particularly in the case of a single story factory building having a large percentage of the wall surface constituted of glass. The outside curtain walls are on the average 42 in. high. The infiltration can only take place on the windward side of the building and, as the infiltration through the wall is an exceedingly small percentage of the total heat loss of building I doubt very much the economic soundness of applying a plaster finish under those conditions. Of course, it is not customary practice.

F. D. MENSING: Professor Larson calls attention to the loss that is likely to result through poor baseboard construction. I think there is a possibility of connection with the *Fire Underwriters*. They are vitally interested in the travel of air from basements up to upper stories. They find a great deal of that is caused by a lack of sealing of vertical construction at the floor lines. I know of many instances where poor construction has happened, and I guess others have also come in contact with it. The Society might make a note of this to make a contact with the *Fire Underwriters'* Laboratories.

E. K. CAMPBELL: It has been pointed out a good many times in the discussions at these meetings that there is apt to be a considerable difference between laboratory conditions and field conditions. Professor Larson has

pointed out the importance of sealing the joints between the wall section and the steel frame, also the importance of sealing the furring at the floor line by means of a baseboard or other means. That has brought to my mind an experience that I had a number of years ago. I was called on a large building that was some 25 or 30 years old. The heating plant was in pretty fair condition but was not giving satisfactory service. The windows were in bad shape, as far as the window construction itself was concerned; but the biggest thing that I did to help out the condition in that building was to put a half a bale of cotton in the joints between the wood frames and the brick due to the shrinkage of the wood frames in that time. So that very frequently in the field conditions you will find a condition that may not have existed when the building was erected, and the importance of it, I think, is pointed out very clearly by Professor Larson's mention of those two things.

H. M. NOBIS: Professor Larson ought to be congratulated on the common sense manner in which he handled a practical subject. I would like to know if it would be possible to ascertain at some time if the wall, after it is plastered, be covered with tinfoil.

C. C. HARTPENCE: Some years ago while building some gas purifying boxes, I decided to build them of brick and I used two coats of neat cement wash on the faces of the 8-in. brick walls. I used cement mortar, two parts of sand and one part of cement, but was very careful to see that the joints were entirely filled. There was no odor of gas in the room and apparently no leakage of gas through the walls of the purifying boxes, though they were under several inches of water pressure. The gas had been cleaned of water and oil mist, but had high relative humidity.

W. C. RANDALL: The point brought out by Mr. Larson that the amount of air coming through the brick wall, particularly if plastered, is negligible, is slightly beside the point in a lot of buildings which are being built today of walls less than 12-in. thick, of a type of workmanship that I think is poorer than the type of workmanship which Professor Larson calls poor workmanship. It is particularly apparent not in the leakage of air but in the leakage of water directly through the face of the building, especially in the multi-story type of buildings. In some of the taller buildings in New York, I find a tendency to go to thinner walls than the 8-in. brick wall, practically only some form of a filler to get an appearance on the outside of the brick. I hope that while Professor Larson has indicated that his tests are about through the discussion that has been made today will stimulate further work along this line.

J. D. CASSELL: In that respect, might I suggest that if you conduct these tests any further you will include wall plastering on the inside, probably waterproofed, and then plastered directly against the brick, not furred. There is a very considerable amount of that character of construction being done, and when the workmanship is poor, as I have had occasion to see in our own work, there will be spots of dampness on a panel, dotted all over, where the moisture drives directly through. A test of that kind would be very beneficial along the line that has been explained here today.

J. E. EMSWILER: I want to congratulate Professor Larson on this very fine piece of work, together with his associates. I know that the paper must represent an enormous amount of tedious experimental work behind it.

S. R. LEWIS: Do not forget the chimney in the outside wall. I was called in on a tall, cooperative apartment building in which they built their fireplace flues into the masonry of the outside wall. The fireplaces served by these chimneys consistently had a backdraft whenever they happened to be on the windward side. The owner installed the biggest fans he could get, with the biggest motors he could put on, up in the attic but could not overcome the trouble. The next apartment like this that I built, I had them build the outside wall; then do things to that outside wall that were paramount to putting plaster on; and then build the chimney inside the wall. We had no trouble due to unauthorized chimney leakage on the windward side when we did this.

G. L. LARSON: I do not know whether I got all these points. Mr. Evans asked some direct questions that I cannot answer. He asked whether the walls were weighed before and after testing. I will say that we did not weigh the walls. The walls were allowed to stand for five months before the tests were made and then check tests were run again two months after the first tests and the difference between the check tests and the previous ones, with an interval of two months between, showed no direct correlation of changes one way or the other. In some cases the leakage was more and in some cases the leakage was less. So we couldn't find any direct correlation there. Apparently, all the drying out had taken place before the tests were made.

We took records, of course, of the wet- and dry-bulb temperatures during all of the tests, but made no attempt whatever to keep these at any constant quantity. They averaged up and down day after day. So we have no relation there. In fact, we could find no relation between the results and the wet- and dry-bulb temperatures that we got during these particular tests. I believe that if the walls were maintained under a certain condition of humidity for certain periods of time we probably would find some difference, but these walls stand there and take the changes as they come and that may vary a great deal over a period of two months.

MR. EVANS: You would then expect more leakage?

PROFESSOR LARSON: Well, I can not say what might happen. Of course, there is one test that we have not been able to perform yet and that is to expose one side of these walls to the outside weather conditions and then test them after a few months of that kind of exposure. We may get into that later but that is not on our program at the present time because we are not equipped to do it.

There was another question on higher wind pressure tests, etc. We have not planned on going to higher wind velocities. In fact the last time we reported on this work, it was suggested we keep down to about 25 mph because it was felt that it was not necessary to go beyond that.

MR. EVANS: That has to do with pent house work?

PROFESSOR LARSON: Yes. I will take it up with our committee and we will be very glad to go to higher wind pressures, if there is a definite request for that. I can see it is of great importance in the work you have mentioned. I also think we can follow out your other suggestion, Mr. Evans, of trying different methods of painting the wall. What we did was just to take an ordinary paint and paint the wall as is done in common practice.

MR. NICHOLLS: One coat?

PROFESSOR LARSON: We tried three coats. A great deal may be brought to light as to how walls should be painted by trying other methods and cleaning out the paint, as you have suggested. We will be very glad to consider that question also.

The other suggestion about waterproofing the wall by Mr. Cassell—I think that is a good suggestion and we are very glad to get it. We want to get any suggestions that will be of any value in bringing out important points in the construction of buildings.

No. 852

EFFECTS OF AIR VELOCITIES ON SURFACE COEFFICIENTS*

By F. B. Rowley, A. B. Algren[1] *and* J. L. Blackshaw,[2] Minneapolis, Minn.

MEMBERS

The results of cooperative research work between the University of Minnesota and the American Society of Heating and Ventilating Engineers

THIS paper, which relates to the effect of air velocities on surface coefficients, is the result of a part of the co-operative research work between the American Society of Heating and Ventilating Engineers and the University of Minnesota. The problem of experimentally determining these coefficients is one which requires the control of several variables, and, also, one which is difficult of exact solution.

The surface coefficient of heat transmission (f) is defined as the number of British thermal units which will flow between one square foot of the surface of the material and the surrounding air per hour per degree difference in temperature between the surface and the air. This definition is direct, but before the coefficients can be determined experimentally there are some of the factors which must be more definitely defined. First, the surface temperature, which is a rather definite quantity, is difficult to determine experimentally, due to the effects of other temperatures. If, for instance, the thermocouples are embedded in the surface, there is the danger that the temperatures recorded will be that of the material below the surface and not of the surface proper. This may be an appreciable factor in materials of low conductivity. If the thermocouples are placed directly on the surface, there is the danger that they will be affected by the air temperature, which may also lead to an appreciable error. After some experimental work, the method selected was to place the thermocouples on the surface of the material and cover them with a thin vellum paper.

The second question which arises in determining the surface coefficient is that of where to take the air temperatures. These temperatures vary with the distance from the surface, and some standard distance must be selected. As will be

* Additional data on this subject will be found in the paper *Surface Conductances As Affected by Air Velocity, Temperature and Character of Surface*, by F. B. Rowley, A. B. Algren and J. L. Blackshaw, p. 429.

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discussed later, temperatures were taken for different air velocities at distances from the surface varying from 0.05 in. up to 4 in. It was found that from 0.5 in. out these temperatures were substantially constant. Therefore, 1.0 in. was selected as the normal distance from the surface for which calculations of surface coefficients were made.

A third point which must be considered is that of air velocities. When we consider the effect of wind on the surface of a material, the question arises

FIG. 1. SURFACE COEFFICIENT TEST APPARATUS AS FIRST SET-UP

as to what angle this wind strikes the surface. It may be parallel to the surface or at any angle up to 90 deg. In any event, it is a question as to how and where to measure the velocity of the air. While this velocity will vary with the distance from the surface, it will also depend upon the direction of the wind. In order to standardize on some practical condition, the velocities were produced parallel to the test surface and measured in this line.

After the test conditions were selected, an apparatus was built up as shown in the photograph of Fig 1 and in the line drawings of Figs. 2 and 3. Fig. 1 shows the apparatus as first set up at the side of the hot box test apparatus. In this case, air at room temperature was used, but it was found impossible to get sufficient variations in mean temperatures between the test surface and the air. For this reason, the apparatus was re-set as shown in Fig. 2. The fan was placed in the cold storage room and the air from the test apparatus was brought back to the room through a return bend. Thus the air temperature

could be controlled for any range, and with reasonable insulation on the pipes there was very little heat loss. With the exception of air temperature control, the two set-ups as shown in Figs. 1 and 2 were substantially the same.

Referring to Figs 1 and 2, the air velocity was furnished by a fan driven by a direct connected 220-volt, variable speed, direct current motor. The air from the fan was delivered to a 6 in. x 12 in. rectangular duct and passed for a distance of 17 ft, at which point a 12 in. x 12 in. opening was provided in the side of the duct for the insertion of the test specimen. From this point,

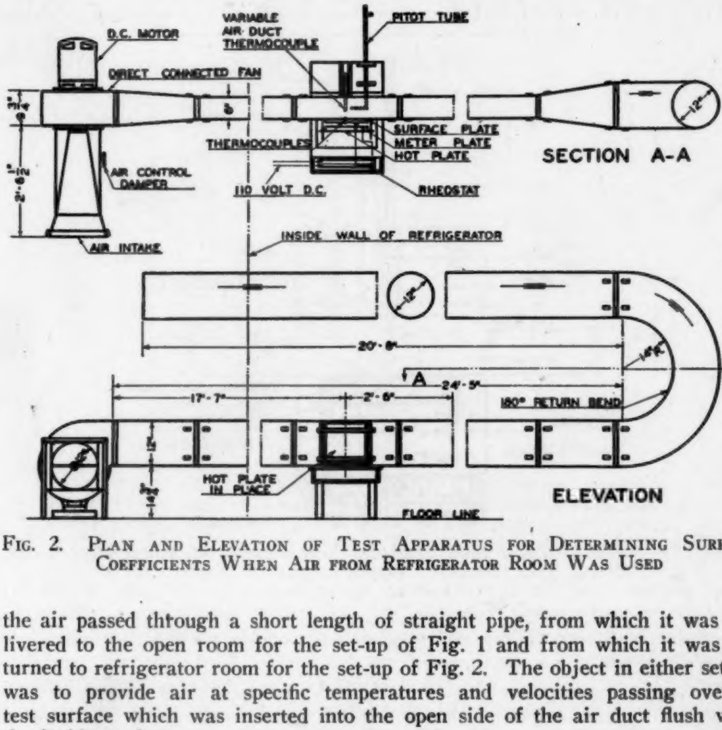

Fig. 2. Plan and Elevation of Test Apparatus for Determining Surface Coefficients When Air from Refrigerator Room Was Used

the air passed through a short length of straight pipe, from which it was delivered to the open room for the set-up of Fig. 1 and from which it was returned to refrigerator room for the set-up of Fig. 2. The object in either set-up was to provide air at specific temperatures and velocities passing over a test surface which was inserted into the open side of the air duct flush with the inside surface.

The arrangement of the test section proper can be best explained by referring to Fig. 3. The test surfaces used were made up of materials 12 in. square and placed in contact with a meter plate which was substantially the same as the Nicholls' heat meter. The meter with the test material in contact was placed in the opening provided in the side of the air duct in such a manner as to bring the test surface of the material just flush with the inside surface of the duct. Thus the air was passed over the test surface in a line parallel to the surface.

The velocity of the air was measured by a pitot tube, which was so arranged

that the velocity could be determined at any distance from the test surface. For low velocities, the Wahlen gage was used, and for the high velocities about 800 fpm an ordinary inclined Ellison draft gage was used. The air temperatures were measured by a fine copper constantan thermocouple, which was arranged to be set at variable distances from the test surface. These temperatures were measured by the use of a potentiometer.

With this apparatus it was possible to get different air temperatures and velocities over the test surface and also to control the temperature of the surface.

The length of straight duct preceding the test surface was sufficient to reduce

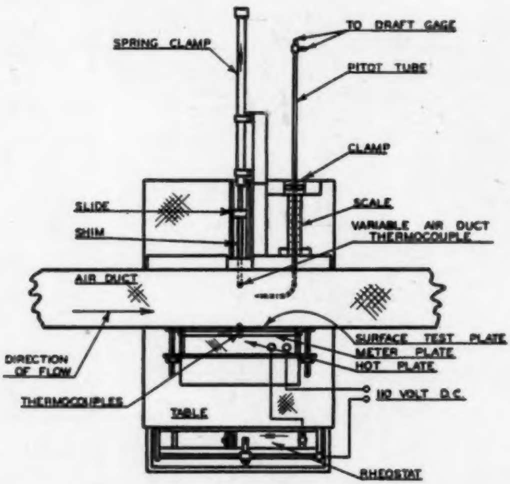

FIG. 3. PLAN VIEW SHOWING ARRANGEMENT OF METER PLATE, TEST SPECIMEN, THERMOCOUPLE AND PITOT TUBE IN RELATION TO AIR DUCT

the turbulence from the fan and to give a reasonably straight line flow over the test surface.

In making the tests, the desired conditions, such as surface temperature, air temperature, and air velocity were first selected and the apparatus was adjusted to give these conditions. It was then allowed to operate for a sufficient length of time to get uniform test conditions and readings were taken for the run. After the temperature drop across the heat meter had been measured, with the aid of the calibration curve for the meter, the amount of heat passing from the test surface could be directly obtained. The surface temperatures were also taken with copper constantan thermocouples and the potentiometer.

The mean temperature was calculated between the air and surface temperatures. The surface conductance was calculated by dividing the number of heat units passing from each square foot of the test surface per hour by the temperature difference between the surface and the air.

There is no specific distance from the test surface at which air temperatures and velocities should be measured. A distance should, however, be selected at which representative results will be obtained under various conditions of wind velocities and mean temperature. In order to determine this distance, many tests were run, using different surface temperatures, air temperatures, and air velocities. In each case, the temperatures were measured at distances from the surface, varying from 0.05 in. to 4 in., and the air velocities were measured at distances varying from 0.125 in. to 5 in. from the test surface. The results for a part of these tests are shown in Figs. 4 and 5. Fig. 4 shows the surface coefficient for a pine surface at zero air velocity and at a mean temperature

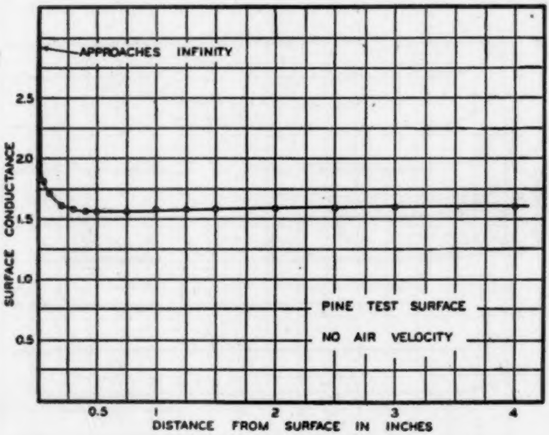

FIG. 4. RELATION BETWEEN SURFACE CONDUCTANCE AND DISTANCE FROM SURFACE AT WHICH AIR TEMPERATURES ARE TAKEN FOR PINE TEST SURFACE AT ZERO AIR VELOCITY AND 65 F MEAN TEMPERATURE

of 65 F. Fig. 5 shows the surface coefficient for the same surface at air velocities varying from zero to 35 mph and mean temperatures of 65 F. In both cases, air temperatures were taken at distances up to 4 in. from the test surface. For tests with moving air, the center of the duct was selected as the proper point for velocity measurements.

An analysis of these curves shows that at a distance of 0.5 in. from the surface, the surface coefficients have reached a constant value. One inch was therefore selected as a reasonable distance at which surface temperatures should be taken for further tests. Fig. 6 shows similar coefficients as determined by the hot box apparatus, with an air velocity of 0.6 mph. With this apparatus, it will be noted that uniform conditions were not reached until about 1½ in. distant from the wall. This difference may be accounted for partially by the unobstructed space in front of the test section. For the hot box apparatus, the nearest obstruction surrounding the test surface was about 10 ft. away, while for the present test apparatus it was only 6 in. away. A further point to be observed in both Figs. 4 and 6 is that as the distance is increased

the coefficient gradually increases. This apparent increase in the coefficient may have been caused by convected air currents acting on the thermocouple.

With the test apparatus as described, surface coefficients were obtained for two surfaces: first, a plaster surface in which the thermocouple was embedded flush with the surface; and second, a pine surface. The pine surface was tested both with a thermocouple embedded and also with a couple placed on the surface and covered with vellum paper. For these surfaces, the effects of wind velocities and mean temperatures were determined. Fig. 7 shows the results which were obtained with the set-up as shown in Fig. 1 and with a plaster test surface having a couple embedded flush with the surface. In this set-up it was difficult to maintain accurately any given mean temperature.

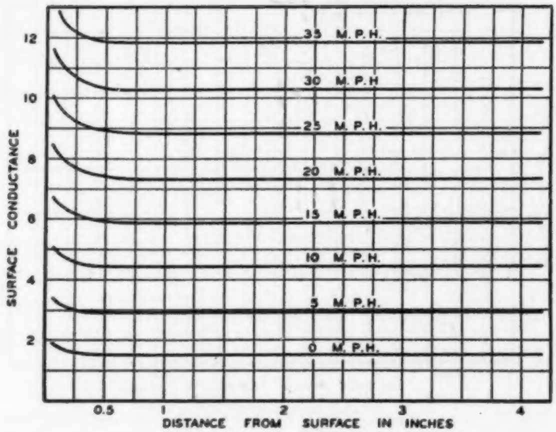

FIG. 5. RELATION BETWEEN SURFACE CONDUCTANCE AND DISTANCE FROM SURFACE AT WHICH AIR TEMPERATURES ARE TAKEN FOR PINE TEST SURFACE AT VARIOUS AIR VELOCITIES AND 65 F MEAN TEMPERATURE

Therefore, the mean temperatures for this set of tests varied from 70 F by plus or minus 10 deg. The points, however, are sufficiently close to establish a straight line relation between air velocity and surface conductance. Fig. 8 shows the results as obtained for a pine test surface with a thermocouple embedded and tested with the same apparatus and at the same maintained mean temperature as for Fig. 7. From the curves of Figs. 7 and 8, it is apparent that there is but very little difference between plaster and pine surfaces, although this cannot be taken as final on account of the variation between the mean temperatures. As previously stated, the lack of definite temperature control in the first set-up led to a second arrangement by which air was supplied from a refrigerator room. The air from this room was controlled thermostatically to plus or minus 1 F. With this apparatus, tests were made with a thermocouple attached to a pine surface and covered with thin vellum paper. For this test the surface and air temperatures were controlled so as to maintain a constant mean temperature between the air and surface of 65 F.

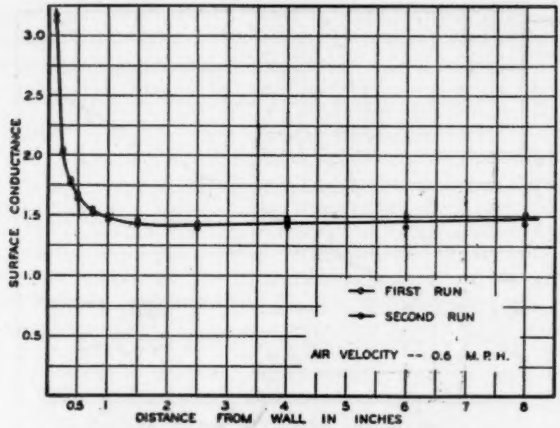

Fig. 6. Relation Between Surface Conductance and Distance from Surface at which Air Temperatures Are Taken. Determined on Hot Box Apparatus

The surface coefficients for this test are shown by the upper curve of Fig. 9. The lower curve is a duplicate of Fig. 8, and, while the average mean temperature for this curve is about 5 deg higher than for the upper curve, it shows a rather definite difference between the coefficients as obtained with thermocouples fastened on with vellum paper and those embedded in the material flush with the surface.

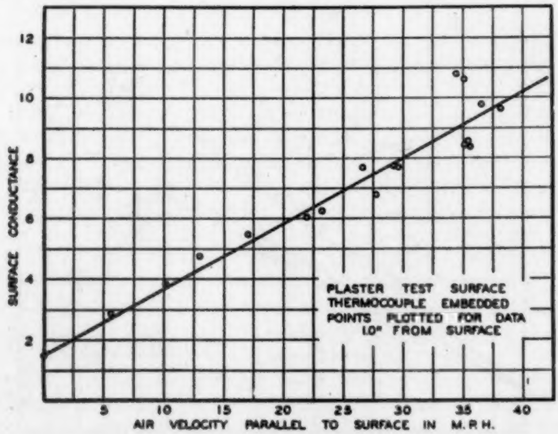

Fig. 7. Relation Between Surface Conductance and Air Velocity One Inch from Plaster Surface and 70 F Mean Temperature. Thermocouple Embedded in Surface

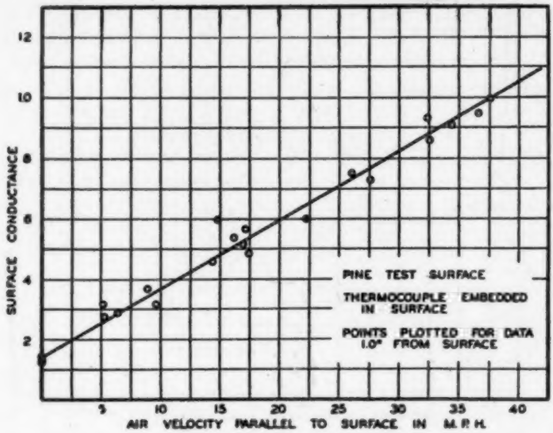

FIG. 8. RELATION BETWEEN SURFACE CONDUCTANCE AND AIR VELOCITY ONE INCH FROM PINE TEST SURFACE—70 F MEAN TEMPERATURE

With the work thus far finished, there seems to be but very little difference between the coefficients obtained for the different types of surfaces. The effects of color of surrounding surfaces have been tried to some extent. Black, gray, and galvanized iron all gave the same results; therefore, a dull gray was chosen for the metal surfaces surrounding the test surfaces.

While the test apparatus as set up gave very uniform results, it was felt

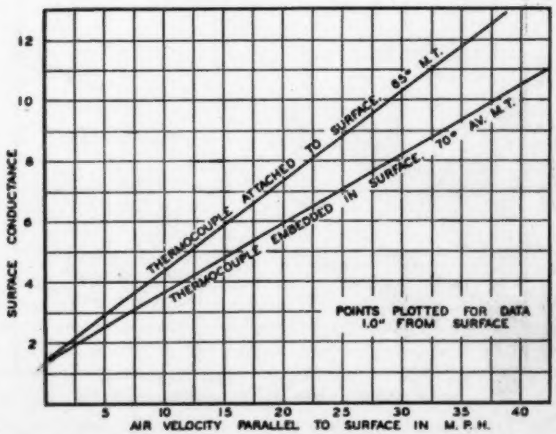

FIG. 9. PINE TEST SURFACE. THERMOCOUPLES ATTACHED TO THE SURFACE AND EMBEDDED FLUSH WITH THE SURFACE

that the meter could be improved. This is, therefore, being rebuilt and the series of tests will be continued to cover other types of surfaces and a wider range of mean temperatures and air velocities. From tests completed, there appears to be some increase in the coefficient with an increase in mean temperatures. This difference is, however, small, and the amount has not been definitely determined for all ranges of temperatures; it will therefore be reported later.

For air velocities ranging from 0 to 35 mph, the coefficients have followed a straight line relation. This line has been steeper for cases in which the couple has been attached to the surface than when it has been embedded. This difference has been due to the fact that the recorded surface temperatures were more nearly equal to the air temperatures when the couples were placed on the surface; therefore, the apparent surface coefficient was higher.

For practical calculations, the upper curve of Fig. 9 may reasonably be used. It will be noted that this crosses the zero air velocity line at a surface constant of 1.34 which agrees with that obtained by Willard and Lichty at the University of Illinois some years ago, and now used in THE GUIDE. The coefficient at 15 miles is, however, somewhat higher.

DISCUSSION

R. M. CONNER (WRITTEN): What effect does the sheet of vellum paper have on the surface temperature measurements? If this strip of paper is placed over the thermocouples, would it not have some effect on the measured temperatures of the surfaces?

If the authors are contemplating making a change in the method of measuring the air passed over the test sample, I would suggest that they investigate the use of a Thomas meter for such measurement.

When large quantities of air are passed over the test sample, is there sufficient heat supplied to that sample to maintain a uniform temperature on the hot side of the test piece?

The statement is made on page 130 of the article that there seems to be but very little difference between the coefficient obtained between the different types of surfaces. This statement seems to be in disagreement with the work which has been done at the University of Illinois, and possibly other work on this subject. The coefficients which we have been using in our test work in connection with warm air furnaces have been taken from the bulletins of the Engineering Experiment Station of the University of Illinois. These constants vary considerably for the different types of materials tested.

PERRY WEST (WRITTEN): It is distinctly understood, I believe, that the data contained in this paper are not supposed to be complete or conclusive. The presentation of the problems involved and some of the indications are very interesting, however, and should be thoroughly discussed here with the full realization that the proper determination of surface coefficients and their application to heat losses is the most important heat loss problem yet to be solved.

The effect of surface coefficients, now in use, varies from about 25 per cent for the average building wall to about 90 per cent for thin metal walls or

glass, so it can be seen that the surface coefficient may be quite an important factor in the total trasmission coefficient.

The transmission data in THE GUIDE 1930 are based on an average still air coefficient of 1.34 for all kinds of building walls and 1.60 for glass with a multiplying factor of 3.0 for walls and 2.40 for glass for a 15-mph wind movement. The still air coefficient of 1.34, for walls, is taken from the average of the coefficients for nine kinds of material, whose surface coefficients range from 0.93 for finished cement plaster to 1.5 for glass.

Taking the highest and lowest of these as the basis for both the inside and outside surface coefficients the variation for the average building wall with still air on the inside and a 15-mph wind on the outside would be 14 per cent, using a multiplying factor of 3.0 for converting the still air coefficients to the 15-mph condition. This factor of 3.0 for converting still air factors to the 15-mph condition is assumed as a fair mean between 3.76 for brick, 2.96 for wood and 2.40 for glass.

If the lowest surface coefficient of 0.93 for cement plaster is assumed to have an air movement conversion factor of 2.40 (same as glass) and the same conversion factor is used on the highest surface coefficient of 1.50 for glass the variation would be about 15 per cent. These extreme variations would show a difference, of course, of from 7 to 7½ per cent from the averages being used.

It may be noted, by reference to the detailed data in THE GUIDE and also by reference to tests by Harding & Willard at the University of Illinois and to tests by the late Prof. A. J. Wood at the University of Pennsylvania, that the character of the surface, as to its smoothness or roughness, has more to do with surface coefficients than does the interior nature of the material itself. This is no doubt due to the fact that a rough surface retards the natural convection currents more than a smooth surface, which means that the still air surface coefficient is lower for rough surfaces than for smooth surfaces due to the fact that the surrounding blanket of air is not changed so rapidly. As wind velocities increase, however, the effect of surface conditions are reduced so that while the still air coefficients are lower the conversion factors are also less, since wind velocities change this blanket just as natural convection currents change it, with a rough surface as compared with a smooth one. These facts are borne out throughout some of the factors now in general use while for others they are the reverse. For example, the still air coefficient for brick work is 1.40 while its conversion factor for a 15-mile wind is 3.76. The same data for glass are 1.60 and 2.40, for wood 1.40 and 2.95 but for hard finish cement plaster which should be more like finished wood or glass they are 0.93 and 2.50.

Comparing some of the data indicated in this paper with corresponding data now in general use, the following is noted:

The still air coefficient indicated for plaster of 1.4 is considerably higher than the coefficient of 0.93 for finished cement plaster taken from Harding and Willard's tests and used in THE GUIDE. It is also higher than the average coefficient of 1.34. The still air coefficient indicated for finished wood of 1.6 is also higher than the coefficient of 1.4 also taken from Harding and Willard and used in THE GUIDE. It will also be found that the coefficients indicated for wind movements are higher generally than those arrived at by using THE GUIDE data.

The factors indicated for wood with a 20-mile wind are from 6.0 to 7.3 whereas The Guide data gives 4.23.

I should like to raise the question as to whether some part of these discrepancies may not have been due to the use of such a small sample of the material being tested, without any guard around the edges, so that a part of the heat transmitted through the meter is not transmitted through to the faces of the sample but out diagonally around the edges, thus showing an undue amount of heat transmitted? Also in the case of wind velocities the measurement of this velocity being at a point 3 in. from the surface may have been closer than the corresponding points at which velocities in former tests were measured. The closer to the sample the velocity is measured the higher the transmission coefficient for any particular velocity should be, as the air movement is naturally decreased as the surface is approached. It is also noted that air currents were employed and measured in a plane parallel to the surface of the test surface. This is in accordance with former tests but raises the question as to how surface coefficients should be used with reference to the exposed surfaces of a building and their positions relative to the direction of prevailing winds. In the case of a square building with one of its sides perpendicular to the direction of prevailing winds, we should have this one side exposed to a wind current at a right angle to it, the two adjacent sides with the wind currents parallel to them and the fourth side in the lea. If this building should be located with one of its corners facing the wind we should have the two adjacent sides exposed to the wind currents at an angle of 45 deg and the other two sides in the lea. In the case where one side is exposed perpendicular to the wind the two adjacent sides would not be exposed to the full velocities of the wind for some distance back from this wall, since the splash of the current at the corners would form pockets, depending upon the velocity. The higher the velocity the greater the unexposed area, up to such a velocity than none of the area of these two sides would be exposed to the wind movement. I think it would be of the utmost importance to know more of these effects of the angularity of the wind as compared with the plane of the surface exposed and as to the effect of splash at the corners.

I would suggest in this connection that a miniature building or box of the material to be tested, in the form more or less of a hot box, with meter plates on the inside against three walls and arranged in, or at the outlet, of a duct so that it could be oriented in reference to the wind directions, might be a good way to test these effects. Recirculation of the air might be arranged for by having an enclosing chamber into which the air is delivered and out of which it is returned to the refrigerator.

This subject brings up again the plan by which exposure factors are worked out by the *Heating and Piping Contractors National Association, i.e.,* to allow one degree of temperature difference for each mile of wind velocity and to apply this to the exposure nearest perpendicular to the direction of prevailing winds and to the two adjacent one eighth points of the compass as well.

From the indication of this paper to the effect that the increments in surface coefficients, due to wind velocity, are a straight line function of this velocity, there is some ground for a direct ratio between miles-per-hour of wind and a temperature difference allowance for same, for glass and thin metal, where practically the entire coefficients are composed of surface effects, provided the

wind velocity maintained on both sides of the surface is within narrow limits. Otherwise, one soon approaches such an absurd condition as is arrived at by allowing one degree per mile for a 70-mile wind with an outside temperature of 70 F which would indicate the same condition as to heat requirements as would be required with zero outside and no wind. The application of such a temperature difference for wind velocity for exposures of any applicable heat transfer resistance, outside of surface coefficients, is, of course, entirely illogical and unscientific.

I believe that this paper has opened up a great many points of this very interesting subject and that it is one of the greatest importance to be followed up to a final conclusion. Surface coefficients for confined spaces such as for air spaces of various characters in composite walls should be studied. Also for the effect that building paper and other membraneous materials have upon the application of surface coefficients both as to their multiplying these surfaces and as to their effect upon the prevention of circulatory air currents which affect the other surface coefficients involved.

F. B. ROWLEY: I do not think that I need to spend very much time in going over the problem of heat transmission. We have reviewed it many times, and I think all of us are familiar with the general aspects of the problem. What we want is to find some reliable method of determining overall coefficients. In general, we are interested in determining heat losses from built-up structures. We know that we can determine the overall coefficients by test methods if we are willing to go to the expense of building up the walls and making the tests, but in practice we would rather determine the coefficients by calculation and not take the time and expense necessary for testing the different combinations of materials. We also know that the theoretical method of determining the overall coefficients by calculation is correct, providing we have the proper coefficients and constants to use in the formulae which have been developed.

In previous investigations, overall coefficients of walls have been determined and checked in many ways. Coefficients for the conductivities of solid homogeneous materials have been obtained and it is felt now by the various experimenters that methods are available by which you can get these coefficients reasonably accurate. It was found that some of the coefficients for air spaces which we were using in the past were not exactly correct. More accurate values have been obtained for some of these and the next problem is that of surface coefficients.

In general, there are three resistances to the heat flow through an ordinary wall. *First,* the surface resistance; *second,* the resistance of the homogeneous materials in the wall; and *third,* the resistance of the various air spaces. This particular part of the investigation relates to the surface coefficients. The work on this problem isn't finished. It is a progress report on a part of the Society's cooperative research program, presented for discussion and criticism.

ALBERT BUENGER: Do I understand from Professor Rowley that he has practically substantiated the data given in THE GUIDE at this time that at a 15-mile wind velocity the surface coefficient is about three times what it is with still air?

PROFESSOR ROWLEY: It is substantially so. It seems a little bit higher. The slope of these lines appears to be a little greater than those used in THE

GUIDE. However, figures in THE GUIDE are certainly safe at present until some more accurate data can be obtained. I think the coefficients for wind velocity are going to show up a little higher for the high velocity than at present used.

JOSEPH PORZELL: Do we understand that the surface coefficients published in THE GUIDE are under some criticism and that the tests under way are endeavoring to determine what coefficients shall be used in the future for publication? Some engineers use surface coefficients slightly higher than published by THE GUIDE and surface resistance normally considered as 0.5. I believe you will find that THE GUIDE represents a figure approximately 40 per cent higher for surface resistance than that commonly used of 0.5.

Another question I should like to ask is, is it not found that oftentimes the laboratory test does not give the accurate result due to very small equipments, such as shown on the screen, due to limited funds? Often a small testing apparatus is used instead of a larger piece of equipment, which would represent more nearly the exact conditions.

L. A. HARDING: I think I can answer in a general way your last statement. The agitation over surface coefficients for testing to determine surface coefficients originated through Professor Willard and myself. We conducted quite a number of tests to determine surface coefficients back in 1915, at the University of Illinois. Our test boxes were not small but were approximately 9 ft in height varying from 3 ft square to 4 ft square. We determined surface coefficients for various air velocities by employing the average velocity of the air through a space 4 in. wide.

MR. PORZELL: Surface resistance.

MR. HARDING: The factor that we are talking about in this discussion I assume is a factor, so-called, of a combination of the surface resistance and radiation and convection. That is what Professor Rowley is talking about and I believe most every one is using a factor somewhere in the neighborhood, for still air, of 1.3. The point, however, is this: at what point are you going to measure this velocity? We should have a standard. It is quite evident that you will obtain a different air velocity at various distances from the surface. One laboratory may measure the velocity half an inch away from the surface and some other laboratory may report coefficients based on velocity measurements taken at some other distance so that the two results are not comparable.

E. R. QUEER: I would like to call attention to the fact that the coefficient that these men have obtained at zero air velocity is about 8 or 9 per cent higher than that given by Willard and Lichty. This is only a small matter, but nevertheless is something to be taken into consideration.

O. W. WALTER: The figures for still and moving air surface conductances which have been used in THE GUIDE in calculating heat transmission constants for walls and ceilings have been checked by a number of investigators who have obtained substantially the same results where the conditions for the tests have been similar. What is really needed now is more corollary information to establish the velocities which should be used for inside and outside air surfaces in calculating heat losses for the actual installation. The velocities for outside surfaces on cold days may need to be higher than 15 mph

in the locality considered; and the air velocity over the inside surface may be considerably higher than zero for average conditions.

Until the present time it has been impossible to get accurate heat loss data from buildings in actual use without the installation of very elaborate test facilities. For this reason very few field tests of this kind have been made. Fortunately, the advent of electric heating opens up new possibilities of obtaining field data on the heat loss from buildings. The heat delivered by an electrical heating system is easily measured monthly or daily by an electric watthour meter, and hourly rates can be obtained by a comparatively low cost graphic demand meter. A Philadelphia concern is assembling data of this character on about 50 installations at this time, and preliminary analyses have already indicated some very interesting results.

Many of the insulated walls used in buildings, where electric heat has been installed, have not been given in The Guide, and it has, therefore, been necessary to calculate the heat transmission constants. In place of the conventional 1.34 and 4.03 used by The Guide, conductances of 2 and 5, respectively, have been used for inside and outside surface conductances. It is believed that these latter figures have been justified by the data which have been taken in the field.

In one particular house six layers of insulation board with air spaces between were used for the walls. The heat losses were not as low as expected. The error in calculation is laid to improper evaluation of the air surface resistances. Even though precaution was taken in installation, it is realized that a chimney effect may have caused air velocities that were not expected in the air spaces. However, it is felt that still air spaces are quite often assumed where they do not exist in actual installations.

A. P. Kratz: To answer the gentleman's question on whether we have still air or not, the term, *still air* has always been more or less misconstrued in connection with the still air coefficients. As a matter of fact, none of them have been still air coefficients; they have been for natural convection.

A year or so ago R. H. Heilman presented a paper to the *A. S. M. E.* in which he calculated built-up equations that separated the effect of radiation and convection under conditions of natural convection. I made some calculations and applied his equations and obtained a very good check for the still air coefficient with the values that we have been using, which are approximately the same as Professor Rowley gives.

Professor Rowley: There is just one thing I would like to say in closing. Somebody has asked whether or not these tests were started with the idea that The Guide was not correct. They were not, and they are proving that The Guide is substantially correct, although not, perhaps, as complete as it might be. Mr. Harding did not tell the whole story. He has designed an apparatus which is in use at the Pittsburgh Laboratory, in which it will be possible to separate the heat transmitted by convection and by radiation. This will bring the coefficients down to a greater refinement. This is perhaps not necessary from a practical point of view, but it is necessary for the purpose of making an analysis and will undoubtedly find practical application.

No. 853

ABSORPTION OF SOLAR RADIATION IN ITS RELATION TO THE TEMPERATURE, COLOR, ANGLE AND OTHER CHARACTERISTICS OF THE ABSORBING SURFACE

By F. C. HOUGHTEN[1] *and* CARL GUTBERLET[2], PITTSBURGH, PA.

MEMBERS

ABSORPTION of solar radiation by buildings is daily becoming of greater importance to the heating and air conditioning engineer. Increase in the application of cooling to buildings in summer is an important factor in this growth of interest. The application of more scientific methods and closer calculation to the estimation of the heating requirements of buildings is also, however, an important factor in the growing interest in this question. With more scientific air conditioning, the relative effect of heat from the sun on heating requirements and proper temperature distribution indoors becomes more noticeable.

The rate of transmission of energy through space from the sun, like the true temperature of the sun, was long a question of controversy among astronomers and physicists. In recent years, however, a close agreement has been reached on both of these values.

According to Bigelow,[3] the radiating isothermal layer or the photosphere of the sun has a temperature of 7,655 C, and emits black body radiation. The radiation leaving this layer would give 5.58 calories per minute per square meter (1295 Btu per square foot per hour) of surface normal to the direction of radiation figured at sea level of the earth. Due to absorption of a part of this energy, while passing through the sun's photosphere, the total radiation of 5.58 calories per square meter per minute figured at sea level is reduced to 3.98 calories figured the same way. Of the 3.98 calories per square meter per minute (880 Btu per square foot per hour) effective at sea level which are received by the extreme outer region of the earth's atmosphere, 2.03 calories are absorbed in passing through to the earth, and only 1.50 calories per square meter per minute (332 Btu per square foot per hour) are actually

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[3] Treatise on the Sun's Radiation, by Bigelow, John Wiley & Sons.

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received at sea level on a clear day. Smoke, haze, fog, and cloudy weather tend to reduce this figure. Determination of the intensity of the sun's radiation at different altitudes gives a total radiation of 2.138 calories per square meter per minute (471 Btu per square foot per hour), at the top of Mt. Whitney, 4420 meters (14,501 ft) above sea level; 1.771 calories (394 Btu), at the top of Mt. Wilson, 1780 meters (5839 ft) above sea level; 1.529 calories (338 Btu), at Washington, 34 meters (111 ft) above sea level; and 1.525 calories (337 Btu) at sea level. Although the total rate of heat transmission through our atmosphere as given above is accurately known, these data are of little or no value in determining the rate of heat absorption by buildings.

In order to establish data on this subject, the Research Laboratory recently made a study[4] of the heat passing through certain types of roof construction

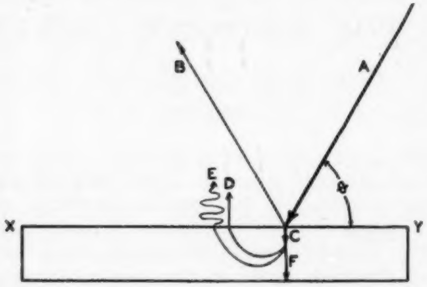

FIG. 1. DISTRIBUTION OF ENERGY OF SUN'S RADIATION UPON STRIKING A SURFACE *XY*

A—Total Radiation. *B*—Reflected Radiation. *C*—Total Absorption by Surface. *D*—Re-radiation from the Surface. *E*—Convection Loss from the Surface. *F*—Resulting Heat Storage in Wall or Made Available for Transmission from Opposite Side.

and the temperature attained by the roof on hot summer days. These observations were made incidental to a general laboratory study of heat transmission through building construction, and no attempt was made to control any of the factors governing the heat transfer. As the sun's radiation falling upon the roof increased in intensity throughout the day, the temperature of the roof rose to a maximum and then receded. In some cases, a high temperature of 140 F was reached. As a result of this study, data were not made available on the rate of heat absorption by a surface for any given temperature conditions.

For a complete understanding of the subject of heat absorption by walls, data should be available giving the rate of absorption by a number of different surfaces for various temperatures of the surface and the air in contact with it. Fig. 1 shows, diagrammatically, what happens when the sun's rays impinge upon the roof or wall surface *XY*. *A* represents the total radiant energy impinging against the surface.

Of this total, a portion *B* does not enter the surface at all, but is reflected

[4] Heat Transfer Through a Roof Under Summer Conditions—F. C. Houghten and C. G. F. Zobel, TRANSACTIONS 1928, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

away from the structure and need not be considered further. The amount of such reflection depends upon the angle Ø, the polish of the surface, and the material of which it is made. In general, metallic surfaces reflect a large portion of the energy rays while such materials as are usually used in building construction, including all but metallic paints, are assumed to reflect a comparatively small portion of the total radiant energy.

The part of the total radiation *A* which is not reflected away from the surface may be represented by *C*. Upon entering the surface the energy changes from radiant energy such as light to heat, and is manifest by a rise in temperature of the material. Due to the rise in temperature of the structure as a result of absorption of the energy *C*, the roof surface *XY* becomes an independent radiator sending radiant energy represented by *D*, back into space—*D* depending upon the character of the surface *XY* and the fourth power of its absolute temperature. Another portion, *E*, of the heat is lost from the surface *XY* by convection currents of air. *E* depends upon the temperature of the air and the surface *XY*, and the velocity of air currents over the surface. The remaining portion of the energy now in the form of heat remains in the structure raising its temperature or is carried away from the opposite surface. It may be represented by *F*.

When we consider all the above phenomena taking place at the same time with different degrees of intensity it is seen that the problem is complicated, and that perhaps the best fundamental data available would be the heat *F* not given back to space or the air on the side of the wall from which the radiation comes. This depends upon the intensity of the original radiation *A*, the reflected portion *B*, which depends upon the angle Ø and the reflecting properties of the surface, the independent radiation from the surface, which depends upon the emissivity of the surface and fourth power of the absolute temperature of the surface, the convection loss *E*, which depends upon air velocity over the surface *XY* and the temperatures of the surface and air. Of these factors the temperature of the surface *XY* and the temperature of the air in contact with it, are perhaps the most important variables, and the data obtained in this investigation were collected with a view of establishing the relationship between *F* and these temperatures for various types of surfaces. Data are also given on the relation of energy absorption to the angle Ø between the direction of the radiation and the plane of the surface, and on the effect of an intervening sheet of glass between the surface and the sun.

Fig. 2 is a drawing, and Fig. 3 a photograph of the apparatus used for collecting the data. The apparatus consisted essentially of a two-foot square surface *D*, made by waxing black oilcloth to a 1/16 in. thick Nicholls heat meter *E*, which in turn was waxed to the copper water cooler *F*. The opposite side of the cooler was insulated by 1 in. of hair felt and 2 in. of cork board. This arrangement made a wall unit 2 ft square by about 5 in. thick which was supported by horizontal and vertical pivots, so that the plane surface studied could be placed at any angle with the direction of the sun's radiation. A skeleton framework was arranged so that a 3-ft square plate of double strength window glass, 1/8 in. thick, could be placed symmetrically between the surface and the sun and 6 in. from the surface. The same skeleton frame work held a paper shield designed to protect the surface studied from wind but not to interfere any more than necessary with natural convection

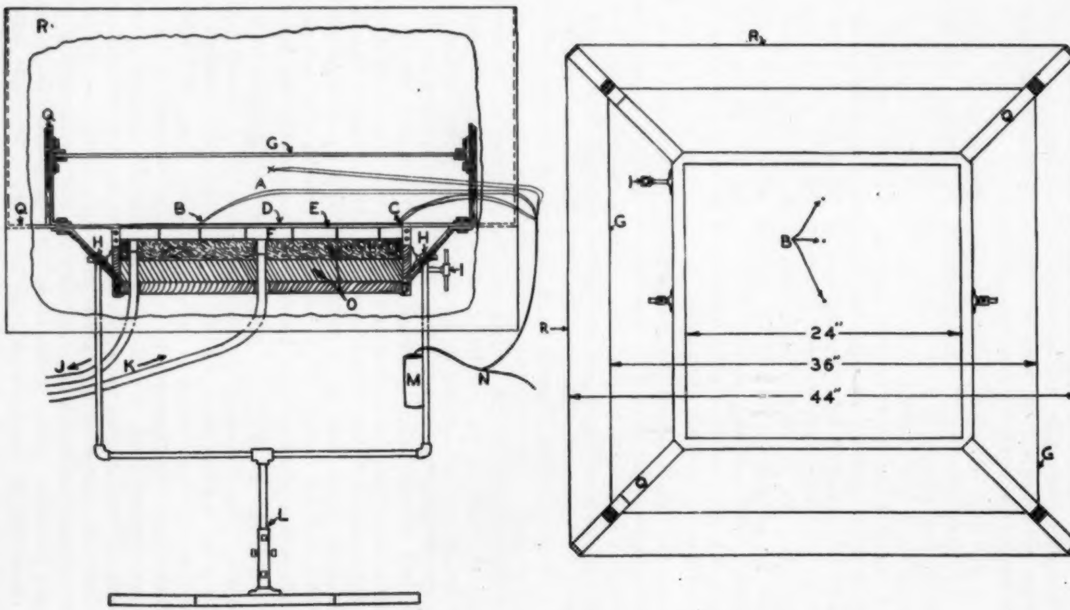

FIG. 2. SET-UP FOR MEASURING RATE OF HEAT ABSORPTION FROM THE SUN

A—Air Temperature Thermocouple. *B*—Surface Temperature Thermocouple. *C*—Heat Meter Leads. *D*—Surface Studied. *E*—Heat Meter. *F*—Water Cooled Plate. *G*—Glass. *H*—Horizontal Pivot. *I*—Sight for Adjusting Angle of Surface. *J*—Water to Circulating Pump and Tank. *K*—Water from Circulating Pump and Tank. *L*—Vertical Pivot. *M*—Cold Junction Bottle. *N*—Cable to Potentiometer. *O*—Insulation, Hair Felt and Cork Board. *P*—Sun Dial. *Q*—Frame for Supporting Paper Shield and Glass. *R*—Paper Shield.

currents. This paper shield in no way shaded the surface studied from the sun's rays.

In order to keep (throughout the test) the desired angle between the plane of the surface studied and the direction of the sun, a piece of ½-in. pipe, 14 in. long was fastened to the wall edge so that its angle with the surface could be set at any desired value; thus when the sun shone straight through this pipe on to a screen below, the surface was in the proper position.

The tests were made on a flat roof of a garage building of the U. S. Bureau

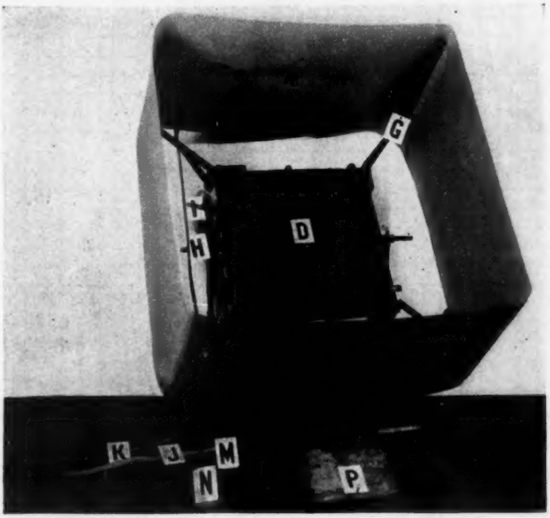

FIG. 3. SET-UP FOR MEASURING RATE OF HEAT ABSORPTION FROM THE SUN. LETTERS ARE THE SAME AS THE REFERENCES ON FIG. 2

of Mines, Pittsburgh Station. This location gave a clear view of the sun from 7:30 a.m. until 6:30 p.m. An insulated and water-proof cable carried the leads from the surface and air thermocouples, and from the heat meter to a potentiometer set-up located in an observation room below. Water, the temperature of which was controlled, was circulated through the cooler in the wall by an electrically driven pump located in this same observation room.

In making a test the desired angle between the surface and the sun's radiation was maintained and water of the proper temperature was circulated through the cooler to give the desired surface temperature. These conditions were maintained while frequent observations were made of the rate of heat flow through the meter and of the surface and air temperatures.

In comparing the relative absorption of the surface for two different conditions of temperature, color, angle, or with intervening glass, tests were made alternately on the two conditions. The thermal resistance between the surface

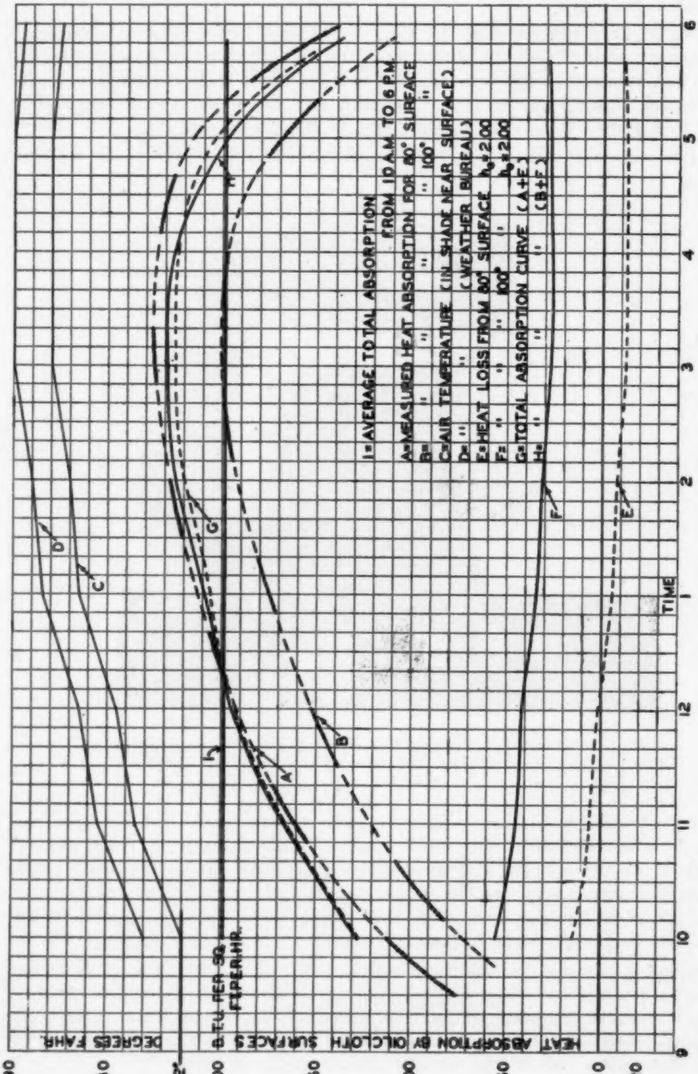

Fig. 4. Heat Absorption of Solar Radiation and Dissipation of Heat by a Black Oilcloth Surface at 80 Deg and 100 F.

TABLE 1. ABSORPTION OF SOLAR RADIATION BY BLACK OILCLOTH SURFACE AT DIFFERENT TEMPERATURES AND DISSIPATION OF HEAT FROM THIS SAME SURFACE TO THE AIR IN CONTACT WITH IT

Time	Btu per Sq Ft per Hour Absorbed F	Surface Temp. t_1	Air Temp. F t_2	Heat Dissipated from Surf. to Air $D+E$	Corrected Heat Absorp. C	Btu per Sq Ft Per Hour Absorbed F	Surface Temp. t_1	Air Temp. t_2	Heat Dissipated from Air to Surf. $D+E$	Corrected Heat Absorption C
Aug. 15 10:10 a. m.	128	54.2	60.0	–11.6	116.4	.....	.....	.....	.....	.....
Aug. 15 1:50 p. m.	194	63.0	62.0	2.0	196.0	.....	.....	.....	.....	.....
Aug. 19 9:45 a. m.	155	71.2	66	10.4	165.4	.....	.....	.....	.....	.....
Aug. 19 11:25 a. m.	215	73.2	68	10.4	225.4	.....	.....	.....	.....	.....
Aug. 20 11:30 a. m.	244	72.5	66	13	257	.....	.....	.....	.....	.....
Aug. 20 12:40 a. m.	280	72.5	76	–7	273	209	102	76	52	261
Aug. 21 11:00 a. m.	175	81	67	28	203	.....	.....	.....	.....	.....
Aug. 21 2:10 p. m.	160	80.5	79.5	28	162	115	100.5	79.5	42	157
Aug. 26* 1 p. m.	37	70	80	–20	17	.....	.....	.....	.....	.....
Aug. 26 2:40 p. m.	175	70	85	–30	145	140	86	85	2	142
Aug. 26 4:10 p. m.	200	70	84	–28	172	.....	.....	.....	.....	.....
Sept. 3 10:00 a. m.	72	100	72	56	128	111	80	72	16	127
Sept. 3 12:13 a. m.	159	100	80	40	199	198	80	80	0	198
Sept. 3 3 p. m.	200	100	86	28	228	237	80	86	–12	225
Sept. 3 5:40 p. m.	128	100	85.4	29.2	157.2	171	80	85.4	10.8	160.2

For F, D, E, see Fig. 1.
$C=F+(D+E)$
$=F+2(t-t_2)$

* Hazy, sun barely visible.

and the water of the cooler was so small that equilibrium of heat flow with any temperature condition usually could be reached in 10 min. Sufficient data on any given condition were always collected to show a steady rate of heat flow for at least ½ hour.

The greatest difficulty was experienced in determining the relative absorption with the different painted surfaces. The paints were made by mixing lamp black, red brick dust, and aluminum bronzing dust with absolute alcohol containing just sufficient shellac to keep the powder from blowing off after it was dry. This paint was found to become apparently dry in four minutes, and satisfactory data could be collected from 10 to 20 min after this time.

Obviously, but one direct comparison could be obtained between the oilcloth and any paint without changing the oilcloth. By adding one paint over the

other several alternate comparisons were made between the lamp black and the red and the aluminum painted surface. Each such application of paint added a slight error due to the fact that the measured temperature was not the true temperature of the surface, but no difference in results because of this factor was noticed.

The effect of temperature of the absorbing surface on heat absorption for the prevailing air temperature on September 3, 1929 is shown in Fig. 4. Similar data for other days are given in Table 1. The points in the solid line portions of each curve represent the measured rate of heat absorption by the surface for the respective surface temperatures. The solid and broken line curves at the top of the chart give respectively the air temperature as measured in a shaded location near the surface studied and as reported by the Pittsburgh Station of the U. S. Weather Bureau about four miles away. It will be noted from the solid line portions of the heat absorption curves that alternate tests covering about ½ hour were made on the two conditions with 10 min or more intervening, which was the time required for bringing the heat flow into equilibrium with the changed temperature conditions.

It will be noted from the air temperature curve *C* that until 12:13 the air temperature was colder than either of the surface temperatures studied. Later in the day the air temperature went up to 86 deg and then remained practically constant. As a result both surfaces were necessarily giving off heat to the air and space until 12:13 noon. At 12:13, the temperature of the 80 deg surface and of the air became the same, and hence no heat was given back to the air by the surface, and necessarily the determination by the meter equaled the total absorption from the sun. After 12:13 the 80 deg surface was cooler than the air and hence it gained heat not only by radiation from the sun but from the air as well.

Considering the small air currents over the surface studied, a surface transmission coefficient $H_o = 2.0$ for the surface studied to the air, or vice versa, may be assumed as reasonable, although we have no accurate data on which to base it. Curves *E* and *F* give the rate of heat exchange between the surface at 80 deg and 100 deg respectively and the air during the period of test. This rate was calculated by the formula $H = h_o\ (t_s - t_a)$

where $H =$ the rate of heat loss from the surface in Btu per square foot per hour.

$h_o =$ the surface conductance coefficient assumed equal to 2.0

$t_s =$ surface temperature

$t_a =$ air temperature.

Curves *E* and *F*, Fig. 4, are represented by *D* and *F* (Fig. 1), combined for the two surface temperatures, while curves *A* and *B* (Fig. 4), for the two temperature conditions are represented by *F*, Fig. 1. It will be noted that the total absorption of radiant energy from the sun by the surface is equal to *D E* and *F* (Fig. 1) combined. Hence the heat transfer indicated by $A + E$ and $B + F$ respectively should represent the total heat absorbed by the surface. Curves *G* and *H* are respectively the algebraic sum of curves $A + E$ and $B + F$. As should be expected *G* and *H* coincide fairly well throughout the period of the tests. Curve *I* gives the average total rate of heat absorption for the period 10 a.m. to 6 p.m.

Table 1 gives the measured heat absorption and the corrected heat absorption for various temperatures and times. It will be noted that for days on which tests were made the maximum corrected absorption was 273 Btu per square feet per hour.

The relative rate of absorption of solar energy for black oilcloth, lamp black and red painted surfaces is given in Fig. 5. The effect of inserting a single

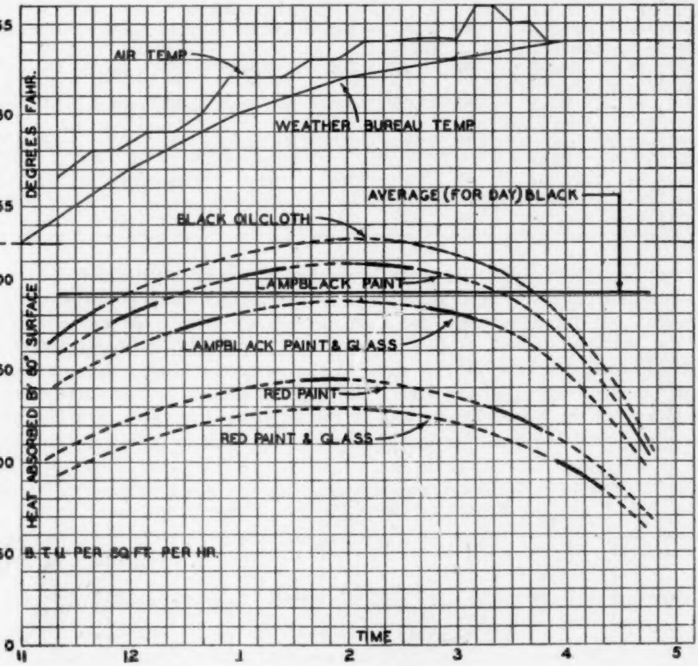

Fig. 5. Absorption of Solar Radiation by Oilcloth and Painted Surfaces With and Without Intervening Glass

pane of double strength glass about 6 in. above the surface through which the sun's rays had to pass is also shown.

Table 2 gives results of other tests on the effect of heat absorption by different surfaces, while Table 3 gives the results of tests on the effect of glass in reducing the rate of heat absorption.

In considering the effect of the glass on heat absorption a number of factors must be taken into consideration. As the sun's rays impinge against, and pass through, an intervening sheet of glass some radiation is reflected directly from each of the two glass surfaces, and additional heat is absorbed in passing through the glass, the amount depending upon the character of the glass and its thickness. All of the heat reflected by the glass surfaces and most

TABLE 2. RELATIVE HEAT ABSORPTION BY DIFFERENT SURFACES

Date and Time	Kind of Surface	Surf. Temp. F	Air Temp. F	Heat Absorp. Btu per Sq ft per Hour	Kind of Surface	Surf. Temp. F	Air Temp. F	Heat Absorption	Heat Absorp. Per Cent. of Blk. Oil Cloth
Sept. 20 11:40 a. m.	Black oil cloth	80	58	182	Lamp black paint	80	58	171	% 94
Sept. 20 1:30 p. m.	Lamp black paint	80	62.3	207	Red paint	80	62.3	143	64.9
Sept. 20 2:50 p. m.	Lamp black paint	80	64.1	203	Red paint	80	64.1	137	63.4
Sept. 20 3:40 p. m.	Lamp black paint	80	65	182	Red paint	80	65	120	62
Sept. 10 12:55 p. m.	Black oil cloth	80	77	256	Aluminum paint	80	77	73	28.5
Sept. 10 2:10 p. m.	Black oil cloth	80	79	222	Aluminum paint	80	79	62	27.9

of that absorbed within the glass is no longer available to reach the surface studied and hence the heat absorption by the surface must be reduced by this amount. At the same time the glass necessarily affects air circulation next to the surface studied, probably reducing it, and also probably effecting a rise in temperature of the air next to the surface. Hence one may assume that the glass decreased the heat given back to the air by the surface studied, or increased the flow when it was in the opposite direction. For this reason the glass would tend to decrease the heat given back to the air by the surface

TABLE 3. EFFECT OF PANE OF DOUBLE STRENGTH (1/8" THICK) GLASS IN ABSORBING RADIANT ENERGY PASSING THROUGH IT

Date	Time	Kind of Surface	Angle	Surf. Temp. F	Air Temp. F	Heat Absorp. Without Glass	Heat Absorp. With Glass	Per Cent Heat Absorbed by Glass
Sept. 15	12:45 a. m.	Black oil cloth	90	80	85	133	120	% 9.8
Sept. 5	2:35 p. m.	Black oil cloth	90	80	90	146	133	8.9
Sept. 5	4:05 p. m.	Black oil cloth	66	80	90	107	96	10.3
Sept. 5	1:45 p. m.	Black oil cloth	57	80	88	136	117	13.8
Sept. 5	3:05 p. m.	Black oil cloth	50	80	90	109	91	16.5
Sept. 5	1:15 p. m.	Black oil cloth	43	80	88	98	82	16.2
Sept. 20	12:30 a. m.	Lamp black paint	90	80	59.3	192	172	10.5
Sept. 20	2:50 p. m.	Lamp black paint	90	80	64.1	203	182	10.5
Sept. 20	4:00 p. m.	Red paint	90	80	64.0	110	98	11

or to increase the apparent absorption. If this is correct the apparent decrease of approximately 10 per cent in absorption by the surface with intervening glass does not represent all the heat intercepted. However, it is doubtful if this factor is very large.

In this connection it is of interest to review some values for absorption of solar energy by glass reported by the U. S. Bureau of Standards. These data were collected in a study of the effectiveness of glass used in goggles for protecting acetylene and arc welders' eyes while at work. The Bureau of Standards values are given in Table 4.

It will be noted from Table 3 that the reduction in absorption of heat by the surfaces studied due to the intervening glass becomes greater as the angle between the surface and direction of impingement falls below 90 deg. This

TABLE 4. ABSORPTION OF SOLAR RADIATION BY VARIOUS KINDS OF GLASS USED IN GOGGLES AS DETERMINED BY THE U. S. BUREAU OF STANDARDS[5]

Color	Trade Name	Thickness in MM.	Transmission Per Cent of Total Solar Radiation
Greenish yellow		2.04	63
Yellow green		2.14	9
Amber		5.57	43
Orange		3.57	47
Sage green		1.95	17
Black	Smoke, C	2.26	60
Neutral tint	Crookes, A	1.97	89
Colorless		1.58	88
Amethyst		2.11	79
Blue-purple		3.13	41
RED		2.90	48
Colorless	Window	1.85	82
Colorless	Crown	1.56	92
Clear	Water	....	76

should be expected since the smaller the angle Ø, or the greater the angle of incidence, the greater the reflection from the glass surface.

The effect of angle of incidence on the absorption of radiant energy is two fold. The larger the angle of incidence or the smaller the angle between the direction of radiation and the surface, the greater the reflection *B*, Fig. 1. For non-reflecting surfaces, however, this factor probably is not very important. More important is the fact that the smaller the angle between the rays and the surface, the lower the intensity of impingement of radiation per unit area of surface. This is purely a geometrical factor and the intensity of impingement per unit of area on any surface is given by $Q=H$ Sine Ø where Q is intensity of radiation falling on the given surface at an angle of Ø with the surface, and H is the intensity of radiation on the same surface normal to the direction of radiation.

The curve (Fig. 6) shows the reduction in heat absorption with the decrease in the angle of incidence. The x's give points for the black oilcloth surface without the intervening glass and the circles give points for tests

[5] Treatise on the Sun's Radiation, by Bigelow, John Wiley and Sons.

with the same surface but with the glass intervening. The curve is drawn for values of Q, given by the equation $Q=H$ Sine $\varnothing$. It is seen that the test points fit the theoretical curve fairly well or in other words for any angle of impingement the formula $Q=H$ Sine $\varnothing$ is apparently the correct form to use for such calculations.

The data given show maximum rates of heat absorption of the sun for different days ranging up to 273 Btu per square foot per hour or up to 1.13

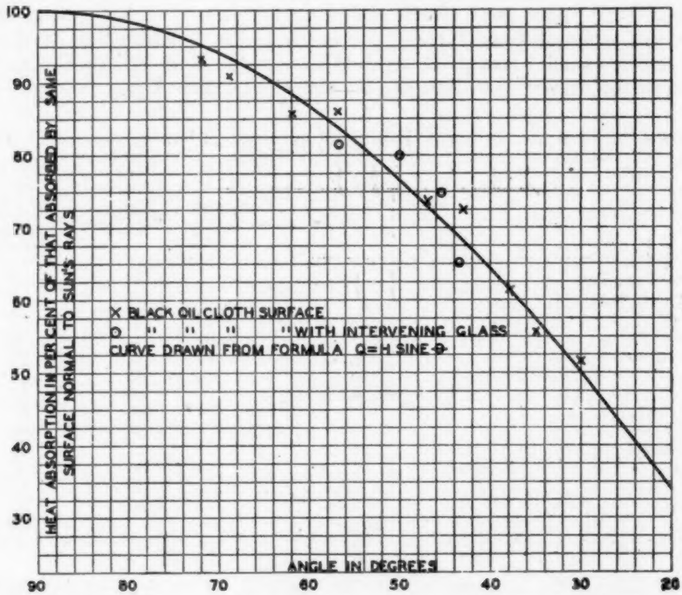

FIG. 6. REDUCTION IN HEAT ABSORPTION DUE TO DECREASE OF ANGLE OF INCIDENCE

sq ft of equivalent steam radiation per square foot. According to Bigelow, the total energy emission from the sun measured at sea level on the earth is about 332 Btu per square foot per hour or 1.38 sq ft of equivalent steam radiation per square foot. This difference can be credited to reflection from the surface studied and lower total radiation from the sun due to smoke and haze while the tests were being made. The tests reported were made on days that would normally be considered quite bright in Pittsburgh. However, the intensity of radiation was probably considerably less than would be found in other places such as Washington or at the sea coast.

SUMMARY AND CONCLUSIONS

1. The total energy emitted by the sun figured normal to the direction of radiation at sea level is given by Bigelow as about 332 Btu per square foot per hour or 1.38 square feet of equivalent steam radiation on a clear day.

2. The heat absorption by a black oilcloth surface perpendicular to the sun's radiation was found to be as high as 273 Btu per square foot per hour on a day which would be considered bright in the city of Pittsburgh.

3. Lamp black painted surfaces, red brick dust painted surfaces and aluminum bronze painted surfaces perpendicular to the sun's radiation show 94.0, 63.4, and 28.2 per cent as high a rate of absorption as the black oilcloth.

4. The difference in retention of absorbed heat by surfaces having temperatures different from the air in contact with them apparently can be satisfactorily corrected for a black surface by assuming a surface transmission coefficient of 2.0 Btu per square foot per hour per deg temperature difference between the surface and air, when this temperature difference is not greater than 20 deg.

5. A single pane of double strength window glass placed so that the sun's rays must pass through it before it impinges on the surface reduces the heat absorption of that surface by from 8.9 to 16.5 per cent, when the impingement is normal. For smaller angles of impingement the glass retards a greater per cent of the radiant energy.

6. The rate of absorption of heat from the sun's radiation by a surface at any angle with the direction of radiation may be satisfactorily computed by the equation $Q=H$ Sine $\varnothing$ where Q is rate of absorption for the angle $\varnothing$ and H is the rate of absorption for the same surface normal to the direction of radiation.

DISCUSSION

HERBERT H. KIMBALL.* (WRITTEN): In this discussion the intensity of solar radiation received at the surface of the earth will be considered first, and afterwards the measurements of absorption by different surfaces.

(1) It is unfortunate that the authors accepted the work of the late Prof. Frank H. Bigelow,[1] as authoritative on solar radiation. By a process of reasoning not generally accepted, he arrived at the conclusion that the so-called solar constant of radiation or the intensity of solar radiation outside the earth's atmosphere at a distance from the sun equal to the mean value of the earth's radius vector, is 3.98 g-cal per minute per square centimeter of surface normal to the incident solar rays. Inside the atmosphere, at the summit of Mount Whitney, or 4,420 meters above sea level, he estimated what he calls the "free heat received by the pyrheliometer" to be 2.138 g-cal per minute per square centimeter. A typographical error in the paper makes it appear that these intensities are for a square meter of surface instead of a square centimeter.

Practically all astronomers and physicists now accept as authoritative the work on solar radiation and atmospheric transmission of Dr. C. G. Abbot, Secretary of the Smithsonian Institution and Director of its Astrophysical Observatory, and his associates. Their results are based on careful measurements extending over many years. A remarkable series of solar constant determinations made at Calama and Mount Montezume, Chile, extends almost without interruption from August, 1918 to the present time. the mean value of these determinations is 1.940 ± 0.0068 g-cal per minute per square centimeter or less than half

* U. S. Weather Bureau, Washington, D. C.
[1] See Bibliography, p. 152.

the value given by Bigelow, which is quoted by the authors of this paper. On Mt. Whitney, pyrheliometric readings evtrapolated to zenithal sun, give for the solar radiation intensity 1.72 g-cal per minute per square centimeter. The authors are quite right in their conclusions that, "These (Bigelow's) data are of little or no value in determining the rate of heat absorption by buildings."

If it were necessary, the intensity of solar radiation at a given place and time could be computed by the use of Abbot's[2] value of the solar constant; Rayleigh's classical equation as modified by King[3] for determining the scattering of solar radiation by the gas molecules of dry air; Fowle's[4] determinations of the scattering by water vapor and the absorption by the gases of the atmosphere, but principally water vapor; and Angstrom's[5] determination of the scattering by atmospheric dust, provided the air pressure and its water vapor and dust content are known. But in the present case it is unnecessary to make such computations, since there are instruments for measuring the intensity of the solar radiation received at the surface of the earth.[6] Not only may we measure the intensity of the radiation received directly from the sun, to which Bigelow refers, but we may also include in the measurement what is almost equally important, that portion of the radiation scattered by the atmosphere that reaches the surface of the earth as diffuse skylight.

Monthly summaries of measurements of the intensity of direct solar radiation made at Washington, D. C., Madison, Wis., and Lincoln, Nebr., are published in each issue of the *Monthly Weather Review. The Review* also contains weekly summaries of the total solar radiation (direct + diffuse) received on a horizontal surface at the three stations named above, and also at stations maintained by the Weather Bureau in Central Park, New York City, at the University of Chicago, and at Fresno, Calif.; at a station of the Bureau of Entomology, Department of Agriculture, at Twin Falls, Idaho, and at the Scripps Institute of Oceanography, University of Southern California, La Jolla, Calif. Beginning with 1930, Pittsburgh, Pa. and the University of Florida, Gainesville, Fla., will probably be added to this list.

In Fig. A are curves showing the annual march of the total solar radiation received on a horizontal surface (*1*) outside the earth's atmosphere at the latitude of Washington; (*2*) and (*3*) on cloudless days at Twin Falls, Idaho, and Washington respectively; (*4*), (*5*) and (*6*), the average daily amount received at Twin Falls, Washington and Chicago respectively; and (*7*), the annual march of temperature at Washington. The relation of (*7*) to (*5*) may be expressed by a mathematical equation (*7*).

These instrumental records, in connection with meterological records, make it possible to obtain a fairly accurate estimate of the solar radiation intensity in any part of the United States.

(*2*) Measurements of absorption by different surfaces represent the contribution of the authors to our knowledge of the effect of solar radiation upon the temperature inside of buildings. Their analysis of the expenditure of the solar radiation (A), received on the roof, namely, B, reflected; C absorbed; D reradiated; E, convection loss; and $F=C-(D+E)$; is similar to the analysis of heat expenditure at the surface of the ground, by Angstrom[7] and others, except that in the latter case a term is added for evaporation of moisture, including the melting of snow and ice. The method of obtaining C,

[2, 3, 4, 5, 6, 7] See Bibliography, p. 152.

however, is quite different. Instead of measuring A, F is measured, D and E are estimated, and C is computed. It would at least be an interesting check to measure A as well as F. Values of D could then be obtained from tables of reflection coefficients.[8]

Table 1 and Fig. 4 indicate that an efficient heat meter was employed in the measurements, since a maximum value of C at 12:40 p.m. on August 20, of 273 Btu = 1.235 g-cal per minute per square centimeter is higher than the average intensity of direct solar radiation at Washington at noon in August, but less than the maximum intensity. The maximum values of G and H

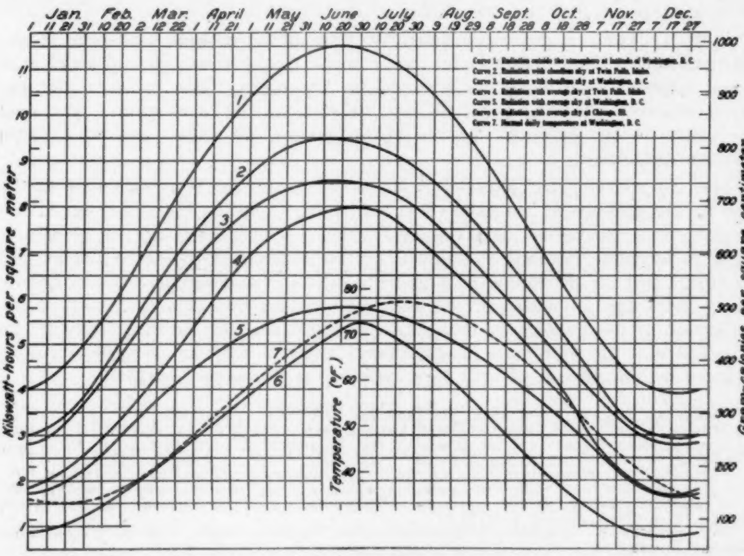

FIG. A. DAILY AVERAGES OF AIR TEMPERATURE AND TOTALS OF SOLAR RADIATION RECEIVED ON A HORIZONTAL SURFACE

(Fig. 4) on September 3 (about 260 Btu = 1.17 g-cal per minute per square centimeter) are only slightly less than we would expect for the value of A in Pittsburgh at noon in September. There is this significant difference, however. The maximum of A should occur at noon, apparent time. The maxima of G and H, at about 3 p.m., or at the time of the maximum temperature of the day. This is in accord with the relation between the curves (5) and (7), Fig. A of this discussion, and with the well-known lag of the daily air temperature behind the mid-day maximum of the solar radiation intensity. It indicates a considerable storage of heat in the roof material.

For the complete determination of the heating effect of solar radiation on a building, the absorption and transmission of heat by its walls as well as by its roof must be considered. From personal experience I may state that an observ-

[8] See Bibliography, p. 152.

atory room with windows on the southeast and southwest sides and warmed by a gas radiator, while uncomfortably cold on a cloudy day in winter is overheated on a day with bright sunshine. It has also been found that in temperate latitudes buildings oriented with their four corners towards the cardinal points of the compass have a more nearly equable distribution of natural lighting and solar heating on all sides, and that the total for the year is greater than for any other orientation.[8]

It appears that attention to the orientation of buildings, to the materials of which they are constructed, and to the character of the paint or other surface covering, will assist materially in the conserving or excluding of solar heat, as circumstances may make desirable.

E. R. QUEER: Some work similar to that of the authors was done at the Pennsylvania State College and the comparative percentages of absorption were practically the same as reported in this paper. For instance, the red paint absorbed about 63 per cent as much heat as the black paint. We used oil paints in our case and the resistance between our meter and the painted surface was higher.

MR. HOUGHTEN: Dr. Kimball of the Weather Bureau takes exception to the use by the authors of the values for total solar radiation from the sun at the earth's outer atmosphere as given by Bigelow. The authors are apparently in error in accepting Bigelow as authority on this subject. It should be pointed out, however, that the discussion of Bigelow's values in connection with the Laboratory data was an academic discussion of theory, and was not used in the collection of the Laboratory data or its analysis, and has no direct bearing on it. There is apparently no disagreement between Bigelow and the Weather Bureau on the subject of total heat received at the earth's surface.

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[1] Bigelow, Frank H. Treatise on the Sun's Radiation. New York, 1918.
[2] Abbot, Charles G. A Group of Solar Changes. *Smithsonian Misc. Col.* Vol. 80, No. 2, 1927.
[3] King, Louis Vessot. On the Scattering and Absorption of Light in Gaseous Media with Application to the Intensity of Sky Radiation. *Phil. Trans. R. Soc.* London, A. 212, 375, 1913.
[4] Fowle, F. E. Water-vapor Transparency to Low Temperature Radiation. *Smithsonian Misc. Col.* Vol. 68, No. 8, 1917. The Transparency of Aqueous Vapor. *Astrophysical Jr.* 42:394, 1915.
[5] Angstrom, Anders. On the Atmospheric Transmission of Solar Radiation and on Dust in the Air, *Geografiska Annaler,* 1929, H.2.
[6] Covert, Roy N. Meteorological Instruments and Apparatus employed by the United States Weather Bureau. *Jr. O. S. A. & R. S. I.* 10:359-369, 1925.
[7] Angstrom, Anders. On Radiation and Climate, *Geografiska Annaler,* 1925, H.1 och 2.
[8] Kimball, Herbert H. The Determination of Daylight Intensity at a Window Opening. *Trans. 1924 I. E. S.,* Vol. 19, p. 217.

No. 854

PREVENTING CONDENSATION ON INTERIOR BUILDING SURFACES

By PAUL D. CLOSE[1], NEW YORK

MEMBER

OF the many problems with which the engineer is confronted, one of the most important is that of preventing condensation on ceilings and other interior surfaces of buildings resulting from the existence of high relative humidities in cold weather. Water dripping from a ceiling may cause irreparable damage to manufactured articles and machinery. It often results in short circuiting of electric power and lighting systems, necessitating shutdowns and incurring costly repairs. Condensation also causes rotting of wood roof structures, corrosion of metal roofs, and spalling and disintegration of gypsum and other types of roof decks not properly protected. There has been much discussion of this subject in the last few years, but to the writer's knowledge, no general formulae have been advanced for the solution of problems of this nature. Charts of various types have been prepared, but these have been of limited value in most cases because they have applied to specific products.

There are two general types of condensation problems, namely (1) those involving the maintenance of high humidities *necessitated* by manufacturing processes, and (2) those involving high humidities which are the *result* of manufacturing processes. In the former case, the relative humidity is usually controlled and has a definitely fixed value. Buildings *requiring* high humidities include textile mills, tobacco curing rooms, film laboratories, bakery dough rooms, match and candy factories, and printing plants. Where the presence of excessive quantities of moisture is the result of some manufacturing process, the relative humidity and temperature usually vary considerably. Buildings involving this condition include paper mills, laundries, macaroni and canning factories, tanneries, stone and marble plants and cleaning and dyeing establishments.

Condensation is, of course, caused by contact of the warm humid air in a building with surfaces below the dew-point temperature. To prevent condensation it is necessary to do one of two things (1) to increase the temperature of the surfaces upon which the precipitation of moisture takes place

[1] Technical Secretary, A. S. H. V. E.

Presented at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1930.

above the dew-point for the dry-bulb temperature and relative humidity involved, or (2) to reduce the humidity of the air so that the dew-point is below the temperature of the surfaces with which it comes in contact. The fundamental upon which the solution of condensation problems is based is that the drop in temperature through any construction is proportional to the heat resistance.

DERIVATION OF FORMULA

In the derivation of a formula for solving condensation problems, the object is to determine the resistance R which a wall or roof must have to maintain the interior surface having a resistance R_s above the dew-point temperature t_d for the dry-bulb temperature t and relative humidity r.h. existing in the building, when the outside temperature is t_o. (See Fig. 1). If t_s is the

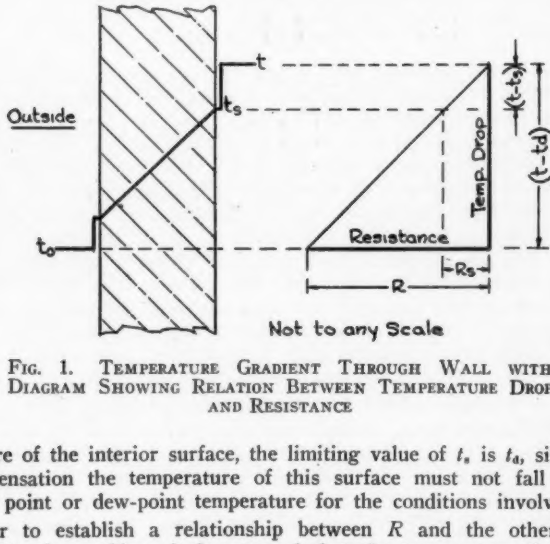


FIG. 1. TEMPERATURE GRADIENT THROUGH WALL WITH DIAGRAM SHOWING RELATION BETWEEN TEMPERATURE DROP AND RESISTANCE

temperature of the interior surface, the limiting value of t_s is t_d, since to prevent condensation the temperature of this surface must not fall below the saturation point or dew-point temperature for the conditions involved.

In order to establish a relationship between R and the other variables entering into the problem, it is assumed that the temperature drop through any part of the wall or roof is proportional to the resistance.

Therefore—

$$\frac{R}{R_s} = \frac{t - t_o}{t - t_s} \text{ and}$$

$$R = \frac{R_s\ (t - t_o)}{t - t_s} \qquad (A)$$

Formula A can be used for solving almost any condensation problem by making $t_d = t_s$, if the necessary data are available. From an inspection of the formula it will be noted that condensation can be prevented in three ways, namely:

(1) By decreasing the surface resistance sufficiently to increase t_s to t_d;

(2) By decreasing the relative humidity so that t_d is equal to or less than t_s, and

(3) By increasing the resistance R so that t_s is increased to t_d.

The surface resistance can be decreased by increasing the velocity of air passing over the surface by a fan, blower or other means. This method is commonly resorted to by store merchants for preventing frost or condensation on store windows, by directing fans against the interior surfaces of the windows.

The second method of preventing condensation, namely, that of decreasing the relative humidity, can, of course, be accomplished by dehumidification.

The third method, that of increasing the resistance of the wall or roof, is accomplished by adding a sufficient thickness of insulation to increase the temperature of the interior surface from t_s to t_d.

If, in formula (A)—

$$t_d = t_s$$

$$\frac{x}{k} + \frac{1}{U} = R$$

$$f = \frac{1}{R_s}$$

where—

$x =$ thickness, in inches, of insulation required to prevent condensation.

$k =$ conductivity of insulation in Btu per hour per square foot per degree Fahrenheit per inch thickness.

$U =$ coefficient of transmission of uninsulated wall or roof in Btu per hour per square foot per degree Fahrenheit.

$f =$ conductance of interior surface of wall or roof in Btu per hour per square foot per degree Fahrenheit, then

$$\frac{x}{k} + \frac{1}{U} = \frac{t - t_o}{f\ (t - t_d)}, \text{ and}$$

$$x = k\left[\frac{t - t_o}{f\ (t - t_d)} - \frac{1}{U}\right] \qquad (B)$$

Example:

The following example will illustrate the use of this formula:

Determine the thickness of insulation required to prevent ceiling condensation on a roof constructed of 1 in. yellow pine sheathing covered with built-up roofing for an inside temperature at the ceiling of 85 F, a relative humidity of 70 per cent, and an outside temperature of —10 F, assuming the conductivity of an insulation to be 0.30 Btu per hour per square foot per degree Fahrenheit per inch thickness.

$k = 0.30$ Btu per hour per square foot per degree Fahrenheit per inch thickness; $t = 85$ F; $t_o = -10$ F; r.h. $= 70$ per cent; $t_d = 74$ F; $f = 1.34$ Btu per hour per square foot per degree Fahrenheit (average); $U = 0.485$ Btu

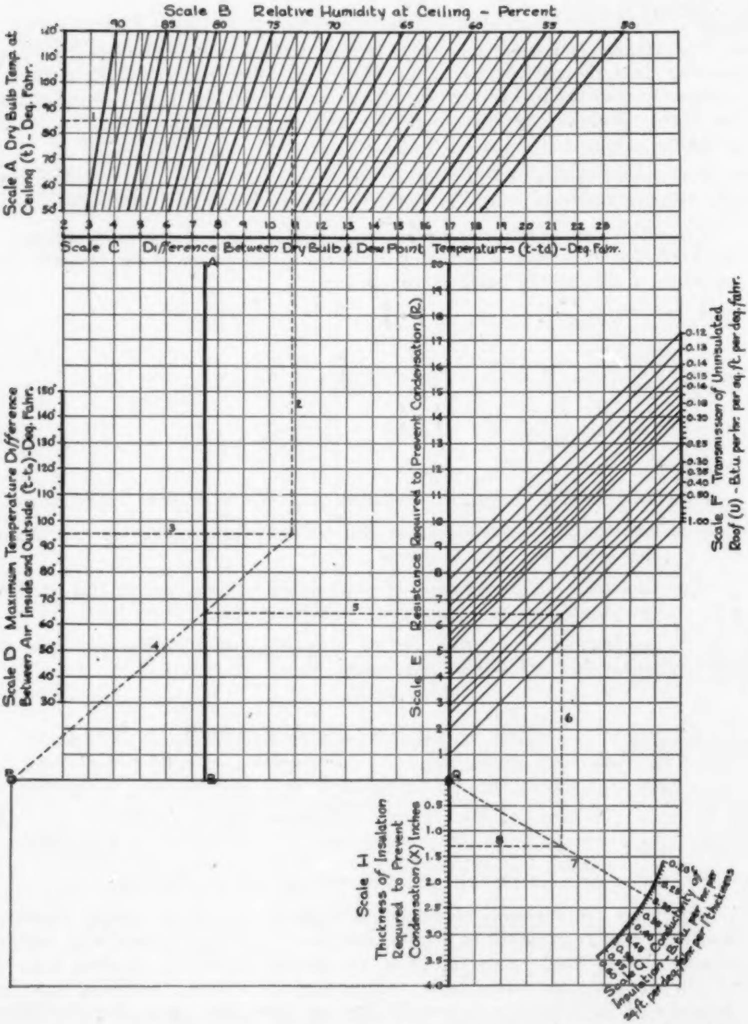


Fig. 2. Chart for Determining Thickness of Insulation Required to Prevent Condensation

per hour per square foot per degree Fahrenheit (Table 11A, Chap. I, The A. S. H. & V. E. Guide, 1929); $x = 0.30 \left[\frac{85-(-10)}{1.34\ (85-74)} - \frac{1}{0.485}\right] = 1.3$ in.

Condensation Chart

In the preparation of the chart (Fig. 2) for ascertaining the thickness of insulation required to prevent condensation, the relative humidity, the inside and outside temperatures, the transmission coefficient of the wall or roof, and the conductivity of the insulation have been considered as variables. The surface coefficient has been assumed to be a constant for still air with a value of 1.34 Btu per hour per square foot per degree Fahrenheit.

The procedure for using this chart is as follows:

1. Locate the dry-bulb temperature t of the air near the ceiling on scale A, and pass horizontally to the proper relative humidity curve on scale B.

2. From this point pass vertically downward.

3. Locate the difference between the temperature of the air at the ceiling and the lowest outside temperature on scale D, and draw a line horizontally until it intersects line 2. (The vertical line drawn as per paragraph 2.)

4. From the intersection of lines 2 and 3 draw a line to the point P in the lower left-hand corner of the chart.

5. From the intersection of line 4 and line AB, draw a line horizontally until it intersects the diagonal line corresponding to the heat transmission coefficient of the uninsulated roof shown on scale F. (See Chap. I, Guide, 1929.)

6. From the intersection found as per paragraph 5, draw a line vertically downward.

7. Locate the conductivity of the insulation to be used (expressed in Btu per hour per square foot per degree Fahrenheit) on scale G and draw a line to point Q.

8. From the intersection of lines 6 and 7, draw a line horizontally to the left, and the correct thickness of insulation to use will be indicated on scale H.

Although this chart is intended primarily for roofs, it can be used for walls by using the dry-bulb temperature and the corresponding relative humidity of the air near the walls at the point which will necessitate the maximum heat resistance to prevent condensation instead of using the temperature and humidity near the ceiling. If the insulation is installed in such a manner as to alter the number of air spaces in the construction, then the total heat resistance required should be determined from scale E from which the heat resistance of the uninsulated wall should be subtracted to ascertain the resistance to be added to prevent condensation. In this way any increase or decrease in the number of air spaces can be taken into consideration by adding or subtracting the resistance of the air spaces to or from the total resistance to be added by the insulation itself. The thickness of the insulation can then be readily determined if its conductivity is known.

Example:

Determine the thickness of insulation required to prevent ceiling condensation for the following conditions:

Dry-bulb temperature near ceiling .... 85 F
Relative humidity .... 70 per cent

Lowest outside temperature..................	—10 F
Construction of Uninsulated Roof..............	1 in. yellow pine sheathing and built-up roofing.
Coefficient of transmission of roof............	0.485
Conductivity of insulation to be used..........	0.30

The solution of this problem is indicated on the chart (Fig. 2) by the dotted line.

1. Locate the inside dry-bulb temperature of 85 F on scale *A,* and draw a line horizontally to the 70 per cent relative humidity curve, indicated on scale *B*.

2. Draw line 2 vertically downward from the intersection located as per paragraph 1.

3. Locate on scale *D* the temperature difference of 95 deg between the ceiling temperature of 85 F and the lowest outside temperature of —10 F, and draw a line horizontally until it intersects with line 2.

4. From the point of intersection of lines 2 and 3, draw a line to the point *P*.

5. From the intersection of lines 4 and *AB,* draw a line horizontally until it intersects with the diagonal line corresponding to a coefficient of transmission of the roof of 0.485, located on scale *F*.

6. From the intersection found as per paragraph 5, draw line 6 vertically downward.

7. Locate the conductivity of 0.30 Btu per hour per square foot per degree Fahrenheit of the insulation on scale *G* and draw a line to point *Q*.

8. From the intersection of lines 6 and 7, draw a line horizontally to scale *H,* on which the thickness of insulation of this conductivity is indicated, which is 1.3 in. The nearest commercial thickness above 1.3 in. would, of course, be selected.

Preventing Condensation for High Humidities

Referring to formula *B,* it will be noted that as the relative humidity increases and approaches 100 per cent, the dew-point temperature approaches the dry-bulb temperature, the quantity $(t - t_d)$ decreases and approaches zero, and the value of x approaches infinity. Theoretically, an infinite thickness of insulation is required when the relative humidity is 100 per cent and the outside temperature is lower than the inside temperature and the quantity $(t - t_o)$ therefore has a positive value.

As the relative humidity increases beyond a certain point, the thickness of insulation increases very rapidly, and a practical limit to the use of insulation is reached. A thickness of 10, 15 or 20 in. of insulation would be out of the question in most cases. The limiting value of the relative humidity for which condensation can be corrected solely by insulation is usually about 90 per cent, but of course, depends on a number of factors, including the construction, the temperature conditions and the type of insulation.

Humidities of 90 per cent or higher are seldom encountered in any manufacturing establishment, because it is usually difficult to approach saturation to this degree under practical operating conditions. Furthermore, humidities this high are considered very unhealthful. However, in instances where such

conditions are maintained and condensation is to be prevented, it can be accomplished in several ways. One method is to install a ceiling a suitable distance from the underside of the roof and to force dry, heated air into the plenum space thus created between the ceiling and the roof deck to maintain the temperature of the under surface of the ceiling above the dew-point temperature. Under the circumstances, the roof should be well insulated as otherwise there will be an excessive loss of heat through the roof structure. Another method is to blow dry, heated air against the ceiling, but this is usually an extravagant and costly process, not only from the standpoint of first cost, but also from the standpoint of maintenance. Moreover, the results are not always satisfactory as it is usually difficult to obtain the proper distribution of the dry, heated air over the ceiling surfaces. The first method is usually to be preferred.

WHERE SHOULD INSULATION BE APPLIED?

From the theoretical standpoint, the most effective results are obtained by applying the insulation to the interior surface of the wall or roof, or as near in the wall or roof to the interior surface as possible, especially if the building is allowed to cool at night and is heated quickly in the morning. The reasons for this will be evident from the following postulates:

(1) The materials nearest the interior surface will have the greatest increase in temperature for any given increase in the inside temperature, if the outside temperature remains constant.

(2) The product of the specific heat, density and volume of the insulation in a given section of the wall or roof is usually less than that of the brick, concrete, stone, tile or other structural materials in the same section.

(3) The absorption of a given quantity of heat by the less dense insulation will raise the temperature of the insulation more than will the absorption of the same amount of heat raise the temperature of the more dense structural materials of the wall or roof.

(4) An insulation will therefore usually increase in temperature more rapidly at a given rate of heat absorption than will the other materials used in the construction.

(5) Consequently, if the insulation is installed on the interior of the wall or roof, there will be less lag between the room temperature and the interior surface temperature during the heating-up period before the temperature of the wall has reached a state of equilibrium.

There are other factors, however, of perhaps even greater importance than the foregoing, which make it advisable to apply the insulation as far as possible from the interior surface of the wall or roof. Probably the most important is that of providing the necessary vapor protection to the insulation, for no insulation will function satisfactorily if it is not properly vaporproofed. All commercial insulations are more or less porous and without adequate surface protection, moisture laden air will penetrate the insulation until the dew-point temperature within the insulation is reached, at which point condensation will take place.

Bituminous emulsions and paints have been used to some extent to protect

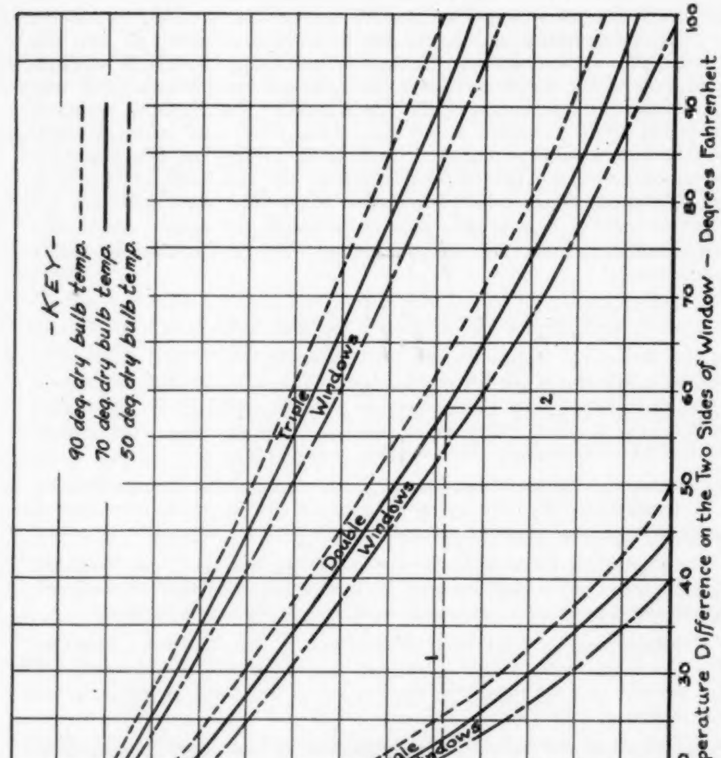


FIG. 3. SATURATION CURVES SHOWING RELATIONSHIP BETWEEN RELATIVE HUMIDITY AND TEMPERATURE DIFFERENCE OR HEAT HEAD FOR SINGLE, DOUBLE AND TRIPLE WINDOWS

insulations applied to the interior surfaces of walls and roofs for the prevention of condensation, but in most cases these have been found wanting. The main reason perhaps is that the irregularities and projections of the surface of the insulation make it difficult to obtain an unbroken seal. Cement plasters have been used with some success for protecting the insulation, but these too

have failed in many cases where severe conditions have been involved, due primarily to the tendency of the plaster to crack as is, of course, true of practically all plasters.

Sheet metal will provide a vapor tight seal if the joints are properly soldered, but certain practical difficulties lessen the value of this material for the purpose. The writer's experience has been that the most satisfactory method of vaporproofing a wall or roof is by means of the membrane system consisting of alternate layers of saturated roofing felt and bitumen applied between the warm humid air of the building and the insulation. Practically all of the insulated roofs of the textile mills of the south are protected in this manner, that is, by the application to the roof deck of several plies of roofing felt and pitch or asphalt before the insulation is "mopped in." Whether the roof deck be of concrete, wood, steel, tile or gypsum, the insulation should be mopped and not nailed to the deck through the roofing felt, if the temperature and humidity conditions are severe, for it is obvious that nails will puncture the vaporproofing course and defeat the purpose for which it is intended. Needless to say, the outside surface of the insulation must be protected from the rain and snow by means of built-up roofing in the case of roofs, and by other suitable methods in the case of walls.

It is frequently the practice to attempt to correct condensation by installing one or more thicknesses of a given type of insulation to the underside of the roof deck, particularly where the roof deck is of wood and the roof rafters thus provide an easy means of attachment. The thickness of the insulation may be sufficient, and the surface of the insulation adequately protected, but in most cases the vapor will penetrate the joints of the insulation even if they are caulked with a suitable plastic, and condensation will take place on the underside of the roof deck, and in turn drop down upon the insulation, rendering it valueless in due course of time. As previously stated, the most satisfactory results are usually obtained where the insulation is applied over a membrane vaporproofing course on top of the roof deck, or similarly protected if applied to walls.

MOISTURE DEPOSITS NOT ALWAYS CONDENSATION

The deposition of moisture on the interior surfaces of buildings due to other causes is frequently attributed to condensation. For example, the formation of moisture on brick, tile and concrete and other masonry walls, often thought to be due to condensation, is quite often caused by the porosity and inherent capacity of such walls for retaining or transmitting moisture. Roof leaks, too, are often mistaken for condensation deposits. In many cases it is difficult to determine whether the precipitation of moisture is due to condensation or to some other cause.

CONDENSATION ON WINDOWS

An interesting paper entitled Frost and Condensation on Windows by L. W. Leonhard and J. A. Grant, was published in the A. S. H. & V. E. TRANSACTIONS, 1929, p. 295. This paper was based on experimental work done in the Mechanical Engineering Laboratory of the University of Michigan to "determine quantitative values for some of the principal factors that influence the inside temperature of windows."

Fundamentally, the problem of preventing condensation on windows is no different than that of preventing condensation on walls or ceilings. Any of the three methods outlined on pages 154 and 155 can be used to prevent or reduce condensation on windows. However, it is not economical to decrease the inside surface resistance as per method (1), since the two surface resistances of a single pane of glass comprise almost the entire resistance of the glass, and therefore any reduction in the resistance of either of the surfaces will result in an appreciable reduction in the overall resistance, and a corresponding increase in the overall transmission. Method (2) involving dehumidification, cannot be resorted to where the high humidity is a necessary part of the manufacturing process. Furthermore, it may not be feasible from the standpoint of the cost of installation and operation of the air conditioning apparatus to attempt to reduce the humidity of the air in the building. It is therefore necessary to resort to method (3) in many instances where it is desirable that window condensation be eliminated, which involves the process of increasing the overall resistance R of the glass. However, the internal heat resistance of glass is practically nil, so that the only means of appreciably increasing the overall heat resistance is by increasing the number of air spaces.

WINDOW CONDENSATION CHART

The chart (Fig. 3) is intended to be used for determining the number of panes of glass required to prevent condensation for certain temperature and humidity conditions, or for determining the outside temperature at which condensation will take place on the inside surface of single, double or triple glass for the humidity and temperature conditions involved. This chart is based on formula (A), assuming still air on the inside glass surface, and a wind exposure of 15 mph on the outside glass surface, using the overall transmission coefficient for glass given in Table 13-A, Chapter I, THE GUIDE, 1929, and the still air surface coefficient for glass of 1.50 Btu per hour per square foot per degree difference in temperature given in Table 4 (Ibid).

The operation of this chart is simple. To determine the type of glass required to prevent condensation, locate the relative humidity on scale A and the difference in temperature between the air on the two sides of the glass on scale B, and the curve corresponding to the inside temperature conditions immediately above the intersection of the lines drawn from these two scales indicates the type of glass required. For example, if the relative humidity is 60 per cent (scale A) and the inside and outside temperatures are 70 F and 0 deg F, respectively, the temperature difference (scale B) will be 70 F, and triple pane glass will therefore be required to prevent condensation as indicated by the curve immediately above the intersection of the lines drawn from scales A and B. It will be noted that curves are given for dry-bulb temperatures of 50, 70 and 90 F, respectively. It is apparent that the difference between the three dry-bulb temperatures is small.

On account of the fact that the transmission of even triple glass is relatively high and, hence, the resistance correspondingly small, the humidity that can be carried without condensation taking place is low, and the greater the difference between the inside and outside temperatures, the lower the humidity that can exist in the building without condensation taking place on the win-

dows. In some cases condensation on vertical surfaces such as walls and windows is not regarded as a serious objection. For this reason, and also because of the impossibility of entirely preventing condensation under severe conditions and the difficulty of hermetically sealing the space between the panes of glass, no effort is made in such cases to reduce or prevent the precipitation of moisture on windows. It is usually the practice, under the circumstances, to provide gutters under the windows to drain the condensation.

The window condensation chart can also be used for determining the outside temperature at which condensation will take place for any given type of glass, and for any specified inside temperature and humidity conditions. If for example, double windows are installed in a building, and the inside temperature and humidity are 70 F and 54 per cent, respectively, condensation will take place when the temperature difference is 58 deg, or the outside temperature is 70 — 58 or 12 F. The dotted lines (marked 1 and 2) on the chart indicate the solution of this problem.

CONCLUSIONS

Condensation on the interior surfaces of buildings is often a serious problem, and one which the heating engineer is frequently called upon to solve. It is caused by the contact of the warm humid air in a building with surfaces below the dew-point temperature, and can be remedied by increasing the temperature of such surfaces above the dew-point temperature or by lowering the humidity.

Dehumidification is not permissible in many cases on account of the fact that a high moisture condition is necessary for manufacturing processes. Hence, the only alternative is to increase the surface temperature, which can be accomplished by decreasing the surface or filament resistance by increasing the velocity of air passing over the surface, or by increasing the overall resistance by adding a sufficient thickness of insulation.

The latter method is generally resorted to, and the thickness of insulation is determined by ascertaining the amount of resistance to be added to increase the temperature of the interior surface above the dew-point temperature for the maximum conditions involved, which in turn is based on the fundamental principle that the drop in temperature is proportional to the resistance.

In order to function satisfactorily the surface of the insulation must be adequately protected from the moisture laden air by means of a suitable vaporproofing course, preferably one consisting of alternate layers of saturated roofing felt and bitumen. It is usually advisable to install the insulation as far from the interior surface of the wall as possible.

Condensation on windows is most readily prevented by using multiple panes of glass with air spaces between, but the heat resistance provided by each air space is relatively small compared with most insulations, so that usually several air spaces are required to entirely prevent condensation under severe conditions. Condensation on vertical surfaces such as walls and windows, especially the latter, is often disregarded, since the most annoying and troublesome problem is usually that of roof or ceiling condensation, to which more attention is paid.

DISCUSSION

R. C. PARLETT (WRITTEN): The author has given a clear presentation of the means for preventing condensation in buildings. Although the principle involved and the calculations necessary in determining the amount of insulation required to prevent condensation are not complicated, the results given in chart form are of considerable value to the engineer who has only occasional need of calculating such a problem. The chart prepared by the author is handy in this respect in that it includes all the factors in the form in which the engineer finds them, and there are no calculations or references to other tables necessary.

Every engineer who has had a number of problems in this field has constructed some chart for his own use, the writer having constructed such a chart for determining the proper thickness of pipe insulation to prevent condensation on cold lines. Tests made by the writer at the University of Wisconsin in 1916, and which check results by other investigators, have not shown surface transmission factors lower than 2 Btu per square foot per degree temperature difference per hour, and the value of 1.34 chosen by the author for his chart based on Harding and Willard's tests appear to be the lowest on record. However, this results in a greater thickness of insulation being used, which is no doubt a good precaution in problems where the only object is to entirely eliminate condensation. The surface transmission coefficient 1.34 is also conservative, because air circulation which increases the coefficient often exists to some degree. While the author's outside surface transmission factor for a roof based on a 15 mph wind is satisfactory for figuring heat losses, the writer has used in condensation problems a surface resistance factor of zero to take care of the action of wind and rain.

The statement made by the author to the effect that it is not possible to entirely correct condensation by means of insulation, where high humidities exist, cannot be stressed too much as insulation engineers are very often called upon to design constructions to take care of these high humidity conditions.

Another point well taken by the author is that for practical reasons insulation is best applied over the roof deck. This generally means that the insulation must be installed at the time the building is erected, and, therefore, should be included in the design by the architect or engineer. In many cases no thought is given to insulation to prevent condensation until after the building is erected and the condition found to exist, at which time it is always expensive and sometimes impractical to apply insulation to correct the condition.

No. 855

STANDARD CODE FOR TESTING AND RATING STEAM UNIT HEATERS[1]

IN order to establish a standard method of testing and rating fan unit heaters using steam as a heating medium, this Code has been prepared by a joint committee of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Industrial Unit Heater Association*.

A. Definitions

1. **Standard Air** is air weighing 0.07488 lb per cubic foot. (This weight corresponds to dry air at 70 F or air with 50 per cent relative humidity at a dry-bulb temperature of 68 F when the barometric pressure is 29.92 in. mercury. Specific heat is taken as 0.2415.)

2. **Static Pressure (S. P.)** is measured at right angles to the direction of air flow.

3. **The Entering Temperature** is the average temperature of the air entering the heater measured at the heater inlet and expressed in degrees Fahrenheit.

4. **Final Temperature** is the average temperature of the air discharged from the heater measured at the heater outlet and expressed in degrees Fahrenheit.

5. **Horsepower** is the brake horsepower input to the fans at a stated speed and stated conditions of rating.

6. **Steam Pressure** is gage pressure in pounds per square inch at 29.92 in. barometric pressure.

7. **Equivalent Direct Radiation (E. D. R.)** is the Btu output at the standard basis of rating divided by 240.

B. Rating

8. **The rating** of a unit heater shall include the following:

A. RPM OF FAN at full load speed

B. HEAT OUTPUT in Btu per hour

C. AT LEAST ONE RATING showing air delivered by the heater in cubic feet per minute of standard air at standard basis of rating

D. BRAKE HORSEPOWER required by fan at standard basis of rating

E. FINAL TEMPERATURE

9. **The rating** shall state the steam pressure (at 29.92 in. barometric pressure) and the entering air temperature at which the Btu and final temperature are taken. It shall also state the temperature at which the cfm value is taken (*i. e.*, 70 F or final temperature or entering temperature).

[1] Code prepared by committee consisting of D. E. French, *Chairman*, O. K. Dyer, G. E. Otis, H. W. Page, W. A. Rowe, J. H. Shrock and L. C. Soule. This committee was assisted in its work by the A.S.H.V.E. Research Laboratory in cooperation with the Mechanical Engineering Laboratory of the University of Kentucky and by the Engineering Committee of the *Industrial Unit Heater Association*.

Adopted by AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *Industrial Unit Heater Association*, January, 1930.

C. Standard Basis of Rating

10. **The standard basis** of rating is to be dry saturated steam at 2-lb gage pressure (at 29.92 in. barometric pressure) at the heater coil and air at 60 F entering the heater when the heater is free of external resistance.

11. **Each rating table** for heaters, for which 2 lb gage is a suitable working pressure, shall include one set of ratings at standard conditions and a statement of its capacity in equivalent direct radiation (E. D. R.).

D. Tolerance

12. **As there** are errors of measurement and inequalities of manufacture, a variation of 2½ per cent in test results will not be considered excessive.

E. Outline of Tests

13. **This Code prescribes** tests to determine the Btu from measurement of condensation by weight and the cfm from condensation and temperature rise, and further prescribes a means for correcting the Btu and temperature rise, as obtained under test conditions, to any other condition of entering air temperature and steam pressure.

14. **It is required that a check** of the air quantity determined from condensation temperature rise be made by direct measurement with a calibrated nozzle. The results of the two methods shall agree within 5 per cent before the tests shall be considered correct. However, the results determined from condensation temperature rise shall govern for the purpose of rating under this Code.

15. **This Code prescribes** that the heater being tested shall discharge into a receiving chamber where the air is thoroughly mixed in order that its true average temperature may be determined. The conditions within this chamber are to be controlled in such a way that the heater delivers the same capacity as when under normal conditions of free delivery.

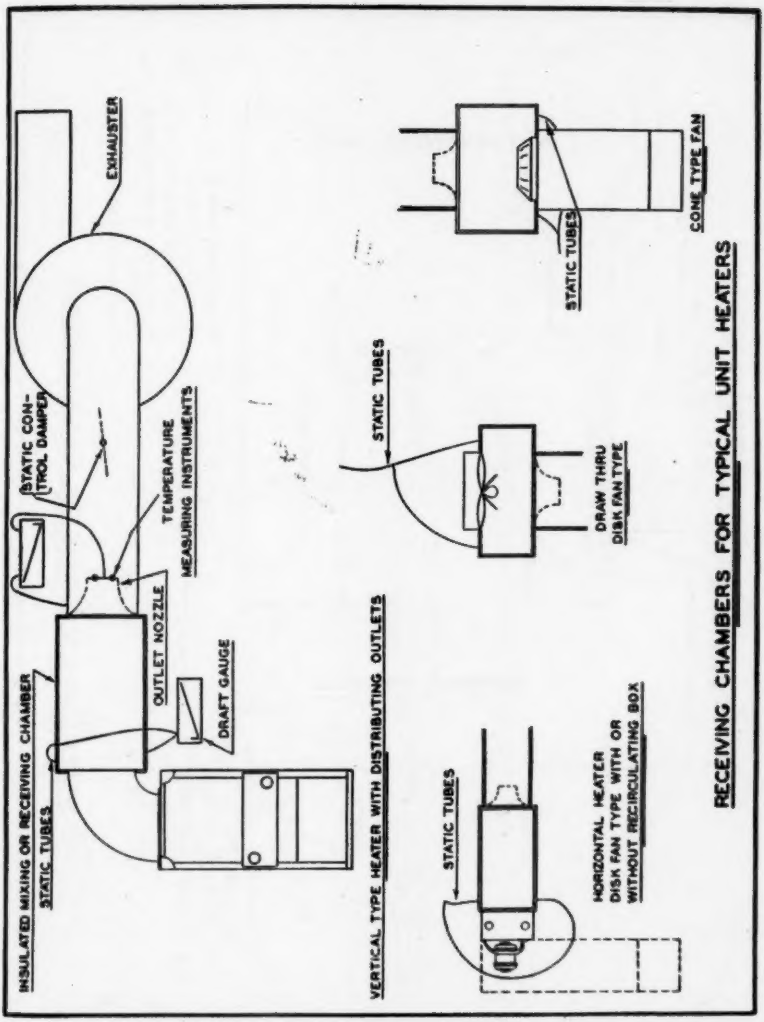
16. **In order** to be sure that the tests on the receiving chamber are a faithful reproduction of free delivery conditions, the following routine is prescribed. First, measure the condensation of the heater during a test under actual free delivery conditions before the heater is connected with the receiving chamber. Then make three tests with the heater connected to the receiving chamber, one while holding—0.05 in., one while holding zero and one while holding +0.05 in. water column constant on the chamber pressure draft gage. During these tests it is necessary to measure only the condensation, entering air temperature and chamber pressure. These may each be of but ½ hour duration. These four tests shall be made at the same fan speed and at entering air temperatures within 4 deg of each other.

17. **The condensation** measured on these three chamber tests shall be corrected to the entering temperature of the free delivery test in the proportion of the temperature differences between the steam and entering air.

18. **A curve** shall then be plotted from the three tests on the chamber showing condensation against chamber pressure. This curve shall be assumed to be a straight line within this narrow range of chamber pressures. From this curve shall be read the chamber pressure corresponding to the condensation measured on the free delivery test. The draft gage, which records the receiving chamber pressure, shall be held constant at the reading thus determined, during the tests necessary for rating data as described hereinafter. This series of tests for the calibration of the receiving chamber to reproduce free delivery conditions shall be made for each heater at each fan speed. Should the free delivery condensation in any series of tests meet the chamber pressure curve for that series at a point above +0.05 or below —0.05 it is indication that too great an error has been made either in the free delivery test or the check tests on the chamber, which must be corrected or the tests repeated.

F. Equipment for Testing

19. **A chamber** for receiving and mixing the air discharged from the heater, to be constructed of any material suitable, to be made air-tight and well insulated (2-in. cork or equivalent).



RECEIVING CHAMBERS FOR TYPICAL UNIT HEATERS

FIG. 1.

20. This receiving chamber shall be connected by a duct to an independent exhaust fan of such capacity that it will overcome the resistance of the chamber and connections and produce a zero static at the point where the heater outlet is joined to the chamber.

21. This receiving chamber shall be of such size that the heater to be tested will produce from 20 to 90 air changes per minute.

22. The outlet or outlets of the heater to be tested shall be joined directly to

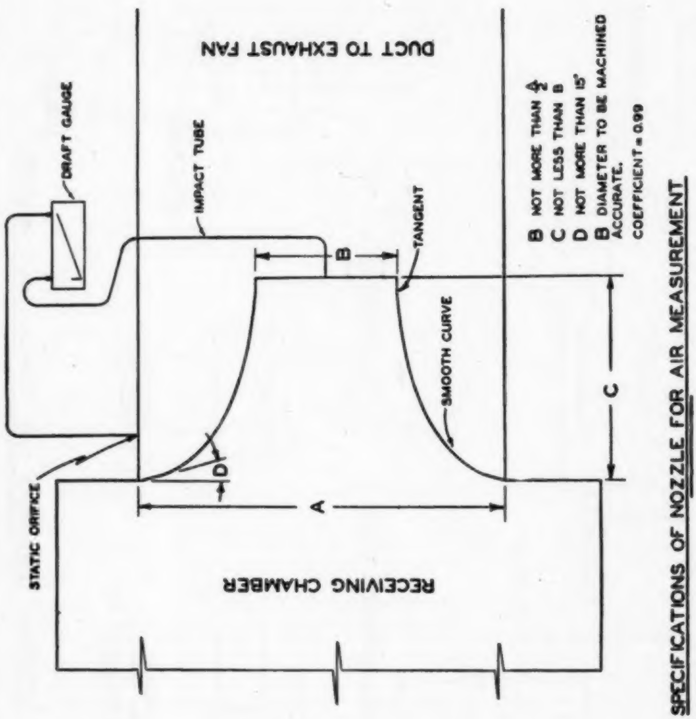


FIG. 2.

the receiving chamber without a communicating duct. (See accompanying sketches of typical cases, Fig. 1.)

23. **Two or more static orifices** shall be located in the receiving chamber and not in the direct air blast from the heater. These openings shall be connected by air-tight tubes to a common draft gage which can be read to 0.005 in.

24. **Means shall be provided** with which to vary the capacity of the exhaust fan as to maintain a zero static constant on the draft gage.

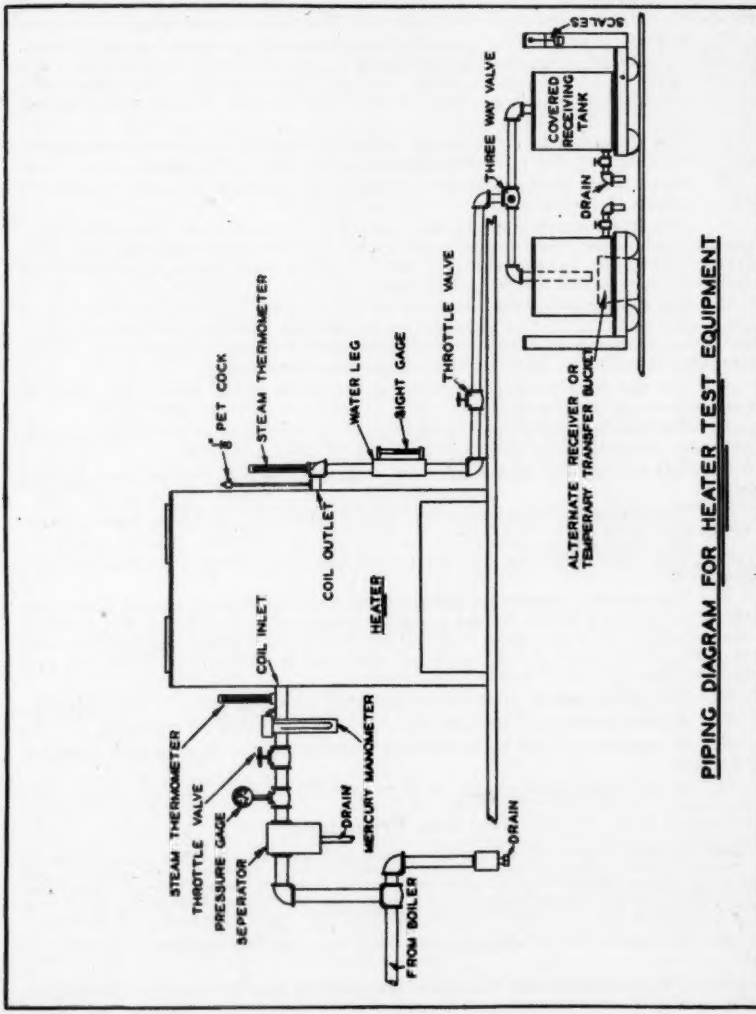


FIG. 3.

25. A calibrated nozzle[3] shall be fitted into one wall of the chamber, discharging into the duct leading to the exhaust fan. The outlet opening of this nozzle shall be of such area that the air velocity is not less than 3000 fpm.

26. Instruments shall be located at the point of 3000 fpm minimum velocity for measuring the final temperature, which shall be the average of temperatures

[3] If the nozzle described in Fig. 2 is used, a coefficient of 0.99 may be assumed without calibration.

taken simultaneously at at least two points in the plane of the nozzle outlet for each square foot of outlet area, but in no case less than four points.

27. **A draft gage** shall be provided for measuring pressures at the nozzle. One side of this draft gage shall be connected to a static orifice located flush with the inner wall of the exhaust duct and near the chamber wall. The other side shall be connected to an impact tube arranged to measure the velocity pressure of the air in the nozzle outlet.

28. **The air handled** by the heater shall be disposed of in such a way as to prevent fluctuation in the temperature of the air entering the heater.

29. **Steam shall be supplied** from a source of sufficient capacity to prevent sudden changes in pressure.

30. **The accompanying piping** diagram (Fig. 3) shows the connections and equipment prescribed for supplying steam to the coil and measuring the condensate. All of the fittings and instruments shown shall be used and shall be installed in the relative positions indicated.

31. **The separator** shall be of liberal capacity. The steam throttle valve shall be of a type suitable for close control. In selecting the type of manometer, it must be remembered that condensation will collect above the mercury on the steam pressure side and lead to error unless compensated for.

32. **The pet cock** used for air relief at the outlet of the heater coil shall be not more than ⅛ in. in size.

33. **The water leg** shall have a sight gage so the water level can be brought to the same point at the time of each reading of the condensate.

34. **The fittings and piping** from the coil outlet to the thermometer shall be insulated.

35. **The scales** for weighing condensate shall be of the beam type, capable of being read to ¼ lb.

36. **The tanks** receiving the condensate shall be covered to reduce loss by evaporation.

37. **Temperature measuring instruments** shall be disposed around the intake of the heater in such locations and in such numbers as will reflect a true average of the entering air temperature. All temperature measuring instruments shall be capable of being read to 0.5 F or closer and shall be calibrated. When exposed to radiant heat they shall be shielded therefrom.

38. **The heater casing** need not be insulated for the purpose of this code.

39. **A stop watch** shall be used for the accurate timing of readings.

40. **A barometer** shall be provided to determine the atmospheric pressure during test.

41. **A revolution counter** shall be provided to determine the rpm of the fans.

G. The Procedure

Note: To avoid duplication, this description deals with the complete rating tests as made on the receiving chamber after the control of the chamber has been checked for reproduction of free delivery conditions. However, such of this procedure as relates to the measurement of entering air temperature and condensation applies equally to the condensation tests under actual free delivery conditions, and at alternate chamber pressures which are first made for this check.

42. **All steam and condensate lines** shall be inspected to make sure they are tight.

43. **The air-tightness of the heater connections** shall be checked, particularly of the insulated chamber up to the point at which temperature readings are taken.

44. **During the test** the steam at the heater inlet shall be held constant at a selected absolute pressure and with a superheat of not less than 2 deg.

45. **Although the test results** must later be converted to a 2-lb basis, it has been found that a higher test pressure, for example 5 lb, provides more uniform test conditions, which is assumed in the accompanying illustrations. This applies to heaters which, when operating at a pressure of 2 lb entering the coil, will have at the coil outlet a temperature higher than the temperature of evaporation. If the resistance to steam flow of the coil is such that steam must be supplied at a

higher pressure in order to accomplish this, the heater shall both be tested and rated at this higher pressure.

46. **Turn on** the steam, open all valves and air vents wide, long enough to blow out all water. Start exhaust fan and heater fan or fans. Set control to give predetermined static in receiving chamber. Set air relief pet cock so that but a thread of steam escapes continuously. Hold steam pressure constant at the coil.

47. **An operator** should be in constant control of the throttle valve, using the manometer as his reference instrument. Before the test is begun a reference line shall be marked on the manometer corresponding to the selected test pressure corrected for the existing barometer and for the head of any water in the manometer. During the test the steam valve shall be manipulated in order to hold the mercury steady at this reference line.

48. **In order to** introduce the required superheat, there shall be not less than 5-lb drop through the throttle valve. If the quality of steam supplied is so low that the required superheat is not introduced by the 5-lb throttling, the line pressure shall be raised until the required superheat is introduced.

49. **Continue warming up** until the air-inlet and air-outlet conditions have stabilized to such an extent that the average temperature change is 1 F or less in 10 min. Check draft gage zero by disconnecting static tube. Reconnect and reset exhaust damper if necessary.

50. **Divert the condensate** to waste tank and determine tare on weighing tank. Be sure that scales are free in operation, and both scales and tank are free from external contacts. Mark on the sight gage of the return trap or water leg the point at which the water level is to be held, at each point of reading the condensate.

51. **Check steam pressure** and temperature, note time to the second, and divert water to weighing tank. This begins the test. The test shall be continued for one hour, during which all conditions shall be held reasonably constant.

52. **The following readings** shall be taken and recorded at intervals of 10 min or less:

A. Steam pressure
B. Steam temperature (coil inlet and outlet)
C. Outlet temperature
D. Inlet temperature
E. Static pressure at heater discharge
F. Weight of condensate for period
G. Rpm of fan
H. Hp input to fan
I. Differential pressure at nozzle.

53. **Exactly at** the end of each period, the condensate shall be diverted to an alternate weighing tank or change bucket. (If two tanks are used for actual weighing, they should each be tared.) The accumulation of condensate for the period shall then be weighed and recorded.

54. **If the data** recorded for successive periods are inconsistent or vary beyond a reasonable margin, the test shall be continued until one hour of consistent data are recorded. Care must be observed to bring the water level in the sight gage to the original position at each reading and to divert the condensate at the end of each reading period to the second.

55. **If the temperature** at the coil outlet falls below the temperature of evaporation for the barometer of test, it indicates either that air is not being removed from the coil or that the resistance of the coil is too high for the selected test pressure.

56. **The horsepower input** to the fans may be computed from the watts input to the fan motor if a calibrated motor is used.

H. Computation of Results

57. **Chamber calibration tests.** The condensation obtained on each of the receiving chamber check tests *C* shall be converted by the following formula to its

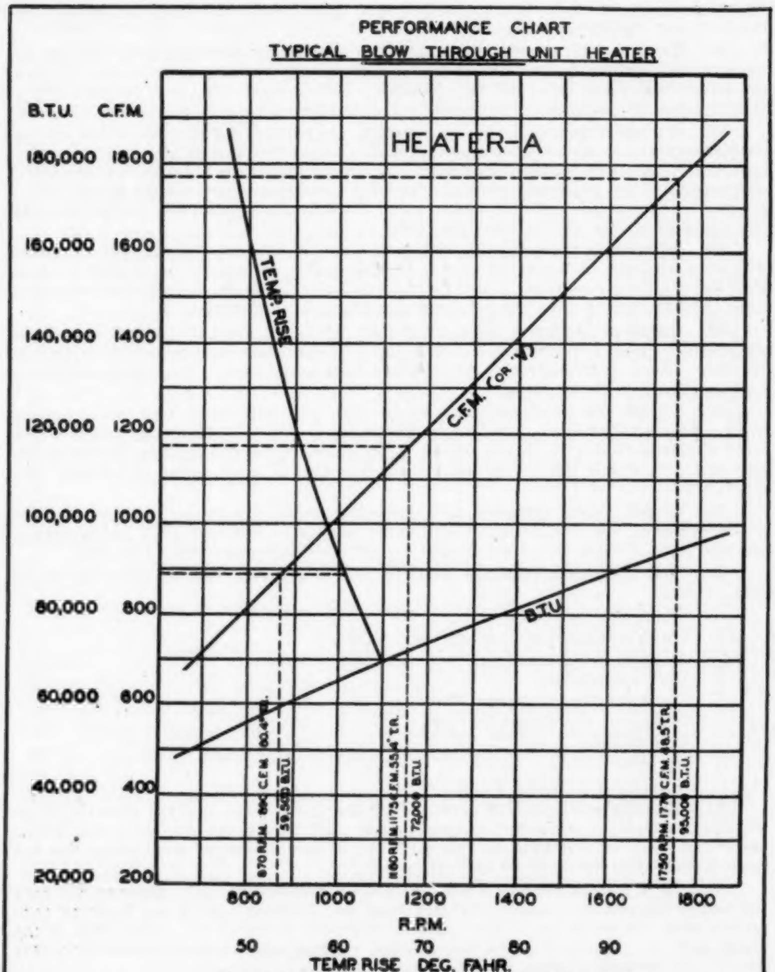


FIG. 4.

approximate equivalent C', at the entering air temperature t'_e of the test made at the same fan speed under free delivery conditions:

$$C' = C\frac{t'_s - t'_e}{t_s - t_e}$$

where t_s and t'_s are the saturated steam temperatures of the receiving chamber test and the free delivery test, respectively, and t_e is the entering air temperature of the receiving chamber test.

58. **Check test of cfm.** The volume of air, in cubic feet per minute, passing

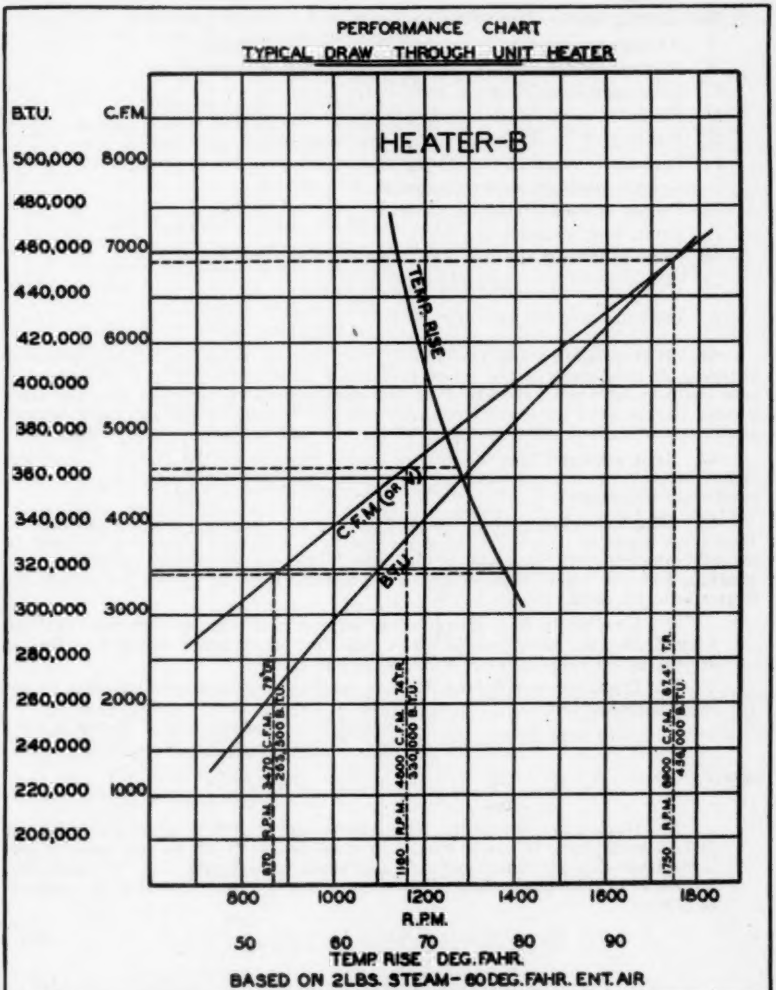


Fig. 5.

through the nozzle, at final temperature as of test conditions, shall be calculated by the following formula:

$$\text{Volume} = 1096\, A \sqrt{\frac{VP}{w}} \times K$$

where A is the outlet area of the nozzle in sq ft, VP is the differential pressure across the nozzle, w is the weight of air at the temperature in the nozzle and the barometer of test, and K is the coefficient of the nozzle.

59. Rating tests. The following result directly from test data:

A. Average temperature of entering air in degrees Fahrenheit, t_e

B. Average temperature of leaving air in degrees Fahrenheit, t_l

C. Temperature rise of air, $t_r = t_l - t_e$

D. Saturated temperature of steam in degrees Fahrenheit, t_s

E. Pressure of steam, pounds per square inch, gage, p

F. Barometric pressure, pounds per square inch, b

G. Total or absolute pressure of steam, $P = p + b$

H. Pounds of condensation per hour, C

I. Latent heat of steam, R.

60. The following are calculated directly from test data:

A. Btu per hour as of test, $H = (C \times R)$ (1)

B. Cfm, standard air, as of test, $M = \dfrac{(55.3 \times R \times C)}{(60 \times t_r)}$ (2)

61. It is generally impracticable to test heaters under exact predetermined entering air temperatures and steam pressures. Since, however, it is necessary to rate them under stated conditions, a standard procedure is given whereby data of test may be used for the determination of performance under any such desired conditions of rating. For this purpose certain basic assumptions are made.

62. It is assumed that the volume of air handled by the fan or fans at the temperature in the fans, is constant for a given heater and fan speed regardless of temperature changes.

63. The following graphic method of solution is prescribed. It requires at least three separate tests on each heater, each at a different fan speed, in order to secure enough points to establish the lines. The speeds selected—one low, one medium and one high—shall embrace the entire range of speeds for which the heater is to be rated.

A. First plot a line, designated as the cfm or V line, as illustrated in Figs. 4 and 5, showing the volume of air in cubic feet per minute V handled by the fan or fans, at the temperature in them, at various speeds.

B. Points for establishing this line shall be computed from test data by the following formulae:

$$\textit{For Blow-Through Heaters, } V = M\frac{(460 + t_e)}{(460 + 70)} \tag{3a}$$

$$\textit{For Draw-Through Heaters, } V = M\frac{(460 + t_l)}{(460 + 70)} \tag{3b}$$

C. Having determined this line, find by means of it at what speeds the fan or fans would have to run to handle an equal weight of air, in each case, at prescribed entering air temperature t'_e and steam temperature t'_s. The equivalent volume V' of this equal weight of air under prescribed conditions shall be computed as follows:

$$\textit{For Blow-Through Heaters, } V' = M\frac{(460 + t'_e)}{(460 + 70)} \tag{4a}$$

$$\textit{For Draw-Through Heaters, } V' = M\frac{(460 + t'_l)}{(460 + 70)} \tag{4b}$$

D. In solving Formula 4*b* it is necessary to determine the final temperature t'_l. Since for the same weight of air the temperature rise is proportional to the differences between the steam and entering air tempratures, the temperature rise t'_r for the desired entering air and steam temperature condition is as follows:

$$t'_r = t_r\frac{(t'_s - t'_e)}{(t_s - t_e)} \text{ and} \tag{5}$$

$$t'_l = t'_r + t'_e$$

E. Next determine the Btu per hour that would have been developed in each

case of test had the same weight of air been handled at other prescribed entering air and steam temperatures, as follows:

$$\text{Btu} = M\frac{(t'_r \times 60)}{55.3} \qquad (6)$$

F. Now determine the Btu line by plotting these Btu values against the speeds (rpm) at which the heater would have to operate, in each corresponding case, to handle an equal weight of air, as of test, at the prescribed entering air and steam temperatures.

G. In order to take off complete rating data at any specified heater speed it is desirable to add a line showing temperature rise, under prescribed entering air and steam temperature conditions, plotted against values of V.

H. For determining the points that establish this line, the temperature rise in each case shall be found by means of Formula 5. These values shall be plotted against corresponding V' values obtained by Formula 4*a* or 4*b*.

I. The capacity of the heater in Btu per hour and the corresponding capacity in cfm of air measured at the temperature in the fan or fans both at prescribed entering air temperature and steam pressure and at any desired fan speed, can then be read directly from the chart.

J. To do this, trace vertically upward along the line representing a selected speed to the points of intersection with the V and Btu lines. Then trace horizontally from these points of intersection to find cfm at temperature of air in fans (cfm) and Btu per hour on their respective scales.

K. To find temperature rise, trace horizontally from the intersection of the speed line and V curve to point of intersection with temperature rise curve. Then drop vertically down to temperature rise scale. This temperature rise plus the prescribed temperature gives the final air temperature.

L. The consistency of the values taken in this manner from the chart should be checked by equating them in the following formulae:

$$\textit{Blow-Through,}\ \text{Btu} = \frac{60(460 + 70)\ \text{cfm}_{\text{fan}}\ t'_r}{55.3(460 + t'_e)}$$

$$\textit{Draw-Through,}\ \text{Btu} = \frac{60(460 + 70)\ \text{cfm}_{\text{fan}}\ t'_r}{55.3(460 + t'_f)}$$

M. If they do not check closely for each selected speed, the computations and plotting and, if necessary, the tests themselves should be carefully looked over for errors. If the discrepancy is negligible, however, the values may be brought into agreement by arbitrary adjustment of the temperature rise.

N. If desired, the cfm at temperature of air in the fan may be converted to cfm of standard air as follows:

$$\textit{Blow-Through,}\ \text{cfm}_{\text{std.}} = \text{cfm}_{\text{fan}}\frac{(460 + 70)}{(460 + t'_e)}$$

$$\textit{Draw-Through,}\ \text{cfm}_{\text{std.}} = \text{cfm}_{\text{fan}}\frac{(460 + 70)}{(460 + t'_f)}$$

SAMPLE DATA—RECEIVING CHAMBER TEST

HEATER A, Blow-Through Type

Speed of fan at test	1772	1177	899
Entering Air Temperature at Test (t_e)	73.4	65.5	75.0
Final Temperature at Test (t_f)	120.3	122.0	132.7
Temperature Rise as of Test (t_r)	46.9	56.5	57.7
Condensation lb per hour as of test (C)	94.5	75.75	59.5
Saturated steam temperature at coil (t_s)	230	230	230
Absolute Pressure (P)	19.7	19.7	19.7
Btu as of test ($C \times R$)	90,700	72,750	57,200
Cfm @ 70 as of test (M)			
$M = \frac{(55.3 \times R \times C)}{(60 \times t_r)} =$	1785	1188	914

Cfm @ fan			
$V = M\frac{(460 + t_e)}{(460 + 70)} =$	1798	1178	922
Temperature Rise @ 60 F Entering 2 lb steam Same weight of air as of test			
$t'_r = t_r \frac{(t'_s - t'_e)}{(t_s - t_e)} =$	48.6	55.7	60.4
Volume at fan with 60 F Entering 2 lb steam Same weight of air as of test			
$V' = M \frac{(460 + t'_e)}{(460 + 70)} =$	1753	1165	897
Btu with 60 F Entering 2 lb steam Same weight of air as of test			
$\text{Btu} = M \frac{(t'_r \times 60)}{55.3} =$	94,100	71,800	59,900

HEATER B, Draw-Through Type

Speed of fans at test	1743	1179	899
Entering Air Temperature at Test (t_e)	86.3	85.8	90.3
Final Temperature—Fan Temperature (t_f)	146.3	151.7	158.4
Temperature Rise as of Test (t_r)	60.0	65.9	68.1
Condensation lb per hour as of test (C)	407.74	302.0	235.75
Saturated steam temperature at coil (t_s)	229.5	229.5	229.5
Absolute Pressure (P)	19.7	19.7	19.7
Btu as of test ($C \times R$)	391,500	290,000	226,000
Cfm @ 70 as of test (M)			
$M = \frac{(55.3 \times R \times C)}{(60 \times t_r)} =$	6030	4060	3060
Cfm @ fans			
$V = M \frac{(460 + t_f)}{(460 + 70)} =$	6900	4690	3570
Temperature Rise @ 60 F Entering 2 lb steam Same weight of air as of test			
$t'_r = t_r \frac{(t'_s - t'_e)}{(t_s - t_e)} =$	67.9	74.2	79.2
Volume at fans with 60 F Entering 2 lb steam Same weight of air as of test			
$V' = M\frac{(460 + t'_f)}{(460 + 70)} =$	6690	4560	3460
Btu with 60 F Entering 2 lb steam Same weight of air as of test			
$\text{Btu} = M\frac{(t'_r \times 60)}{55.3} =$	444,500	327,000	263,500

Note: In writing this Code, the committee had in mind the need for commercial simplicity and practicability, eliminating from the test procedure and computations certain complications which, while they would promote greater accuracy, are not, however, essential in producing results that are accurate within the tolerances established for this Code. These are, however, discussed in the addenda.

ADDENDA

Standard Code for Testing and Rating Steam Unit Heaters

A1. These addenda include

A—Recommendations for the construction and use of the equipment prescribed by the Code, discussed in more detail than was feasible in the Code itself.

B—Discussion of refinements by which the Code test procedure can be elaborated by those interested in a higher degree of accuracy than the Code requires.

Recommendations—Code Equipment and Its Use

A2. A receiving chamber is prescribed by the Code in order to effect a thorough mixing of the air discharged from the heater so that its true average temperature can be read accurately, but the conditions in this chamber shall be controlled in such a way that the normal free delivery conditions of heater operation are duplicated.

A3. Even though insulated, this chamber shall be relatively small, so the heat loss through its walls, which is neglected by the Code, will be a minimum. This leads to the specification in the Code of 20 to 90 air changes per minute. Yet the chamber must be large enough so that the velocity of the discharged air is reduced to 40 per cent (or less) of the outlet velocity through the net outlet area of the heater. This is desirable for accurate control of the pressure in the receiving chamber to produce free delivery conditions. It is further desirable that the discharge air pass from the heater outlet to the point where its temperature is measured in less than three seconds.

A4. With these requirements in mind the size and proportions of a suitable mixing chamber can be readily designed for a given size of heater. However, the manufacturer who has a number of sizes to be tested may want to provide a chamber, the proportions of which can be readily adjusted rather than provide two or more chambers for his range of heater sizes. For this purpose it is recommended that the chamber be designed in the proportions that are correct for the largest heater, but that it be provided with two insulated partitions to be placed parallel with the air flow so that the width of a central passage through the chamber can be varied to suit the requirements of the smaller heaters.

A5. Slides to receive these partitions can be built into the chamber so that the partitions can be moved from one to the other readily as required. An adapter plate will be required for each size heater to connect that heater to the fixed opening of the chamber. The openings in this adapter plate to receive the heater outlets should be so located that the heater outlet or outlets are centered in the receiving chamber opening.

A6. When the heater is operating under normal conditions of free delivery, there is a zero static pressure around the periphery of the heater outlet. This is the condition to be reproduced within the receiving chamber. In order to give a true reflection of this condition, the static orifices should be located in or near the plane of the heater outlet, and within 6 in. of the periphery of the heater outlet and not in the path of the air, i. e., not at a position where the static orifice can register velocity pressure.

A7. Orifice plugs set into the walls of the chamber with their faces flush with the interior surface of the walls, have been found to give good results. For housed fan heaters or disc fan blow-through heaters, these orifices should be located in those walls of the chamber that are parallel to the flow of the air as it passes through the chamber. For disc or cone type fan draw-through heaters, where the fan itself is inside the chamber and has a tendency to throw off air radially, the orifices should be located in the end wall which is in or back of the plane of the heater outlet. No more exact requirements for the positioning of these orifices can be given in view of the wide range of the types of heaters in common use.

A8. While it is correct to assume that free delivery conditions are reproduced if zero static is maintained around the periphery of the outlets when the heater is connected to the chamber, it must be remembered that unlooked-for eddy currents within the chamber might influence the static orifices, no matter how well their location is selected, so that they would give a false reading of the true static condition. To guard against error from this source, a series of check tests is provided in the Code, so that the rating tests can be run at that reading of the static orifice draft gage at which the free delivery condensation of the heater is found to be duplicated. This reading might not be zero, but it should approach it. The fact that it may not be zero is accounted for by the possible influence of eddy currents or by the presence of unavoidable error in the check tests from which this reading was determined.

A9. If the reading indicated by the check tests as reproductive of free delivery conditions exceeds + or —0.05 in. water column, it indicates either that errors in the check tests are greater than allowable or that some unexpected condition within the chamber has an undue influence on the static orifices. If this occurs, a repetition of the tests usually shows that the former is the cause. It may be necessary, however, to re-position the static orifices.

A10. Since each size of heater at each fan speed produces a different condition within the chamber, it is correct to repeat the check tests for each fan speed of each heater and the Code so provides when rating tests are to be made.

A11. Once the position of the static orifices is approved through check tests on one size of heater, it will be found that pressure readings indicated by check tests on other sizes of heaters will fall within a narrow range above and below zero. If so, check tests may be omitted from routine tests (not rating tests), without objectionable error, holding zero static uniformly throughout.

A12. The Code requires a check measurement of air quantity by means of a nozzle. Any form of nozzle or orifice may be used if it is properly calibrated but the form described in the Code is recommended and, if made to Code specifications, need not be calibrated.

A13. While 3000 fpm is the minimum for the velocity of the air at the nozzle outlet, there is no maximum limit to this velocity except the practical one of the pressure required of the exhaust fan to overcome the high resistance of the nozzle at extreme velocities. If the exhaust fan has sufficient capacity to handle the air from the largest heater against the resistance of chamber and duct, plus a resistance of approximately 2¼ in. for the nozzle, only two nozzles need be provided for each chamber, one of one size for the first part of the range of the chamber and one of another size for the second part of the range. If desired to use an exhaust fan of lower pressure, one or more additional sizes of nozzle can be provided.

A14. It is important to prevent fluctuation in the temperature of the air entering the heater during the test and to prevent undue stratification of the air entering a heater having its inlet in a vertical plane. The size and arrangement of test rooms varies so widely that no specific recommendations are possible, but in general it is well to arrange the discharge from the exhaust fan so that most of the heated air is discharged out of doors or into a room that does not communicate with the room from which the heater draws its air. It is well also to admit outside air to the heater through openings so large that their resistance is negligible, as outside temperatures vary but slowly and stratification will be a minimum.

A15. In selecting the type of manometer, it must be remembered that condensation will collect above the mercury on the steam pressure side and lead to error unless compensated for. It is recommended that the manometer be so constructed that a head of water is maintained above the mercury on the steam-pressure side, so arranged that its level will be maintained constant by means of an overflow return to the steam manifold. This water column should be confined in a chamber having a liberal cross sectional area by comparison with that of the manometer tube, so that changes in the level of the mercury column will affect the water level but little. The overflow level should be an accurately measurable distance from the scale of the manometer for accurate calibration.

A16. Where Thermocouples are used as temperature measuring instruments, the cold junction should be immersed in a medium kept at the temperature of melting ice, in order to increase the differential between the hot and cold junctions and thus provide for greater accuracy.

A17. Where heaters of larger capacity are being tested, the condensate in the weighing tanks accumulates with some rapidity and it is therefore advisable to provide weighing tanks with a rapid emptying device so that the condensate of one receiver can be disposed of before the other receiver is filled.

A18. For the check test, in order to produce free delivery conditions conveniently and, at the same time, to remove from the test room the heated air discharged from the heater, it is suggested that the heater be set up with the heater outlets on a level with and directed toward the receiving chamber opening but separated from it by a distance of not less than 3 ft. Thus the air delivery of the heater will not be influenced either by the proximity of the receiving chamber or the operation of the exhaust fan which should be run at a capacity somewhat greater than the air capacity of the heater. Also the heater can then be moved into position for attachment to the receiving chamber with a minimum of work, particularly if the piping connections are arranged with this in mind.

Provision and Corrections for Greater Accuracy

A19. Some of the water condensed by the unit during test escapes through evaporation from the receiver. This loss is neglected in the Code after being minimized by covering the receiver. It may further be reduced by providing a cooling device in the return line to the receiver. Any error through loss by evaporation tends to give Btu and cfm ratings which are less than actual.

A20. The Code does not require a correction for humidity. If the humidity at the time of the test is lower than standard and no correction is made, the Btu rating of the heater will be less than its actual capacity when the enering air is standard humidity and *vice versa.*

A21. The Code prescribes that the air relief valve shall be slightly cracked to allow a thread of steam to escape in order to make sure that the coil is kept free of air. It would be possible, if any appreciable amount of steam escaped in this way, for this steam to carry with it a slight amount of water that would otherwise increase the weight of condensate chargeable to the heater. If a small pet cock is installed so that but a thread of steam escapes, it is considered that error from this source is negligible. Any such error is in the direction of low Btu and cfm ratings. Any such error might be avoided by substituting a thermostatic air valve for the pet cock, but it is then necessary to watch more closely for evidence of air binding.

A22. The Code neglects the heat added to the air by the power expended to move the air through the heater. This makes for low Btu and cfm ratings. For the purpose of the Code, this error is neglected because it is so small in magnitude and because it partially or completely offsets loss by radiation through the heater casing, which is opposite in sign in its effect on the air quantity rating, and which is also neglected in the Code.

A23. For extreme accuracy the heater casing should be insulated and a further correction made for the heat loss through this insulation. Any unaccounted for loss from this source results in a higher indicated air quantity rating than actually delivered. If correction is made for this loss a correction should also be made for power expended to move the air.

A24. In some heaters there may be a cooling of the condensate in the coil appreciably below the temperature corresponding to the pressure at the coil inlet. This transfer of heat from the liquid is not credited to the heater in the Code procedure which credits only the latent heat. Also the Code method neglects any heat of superheat.

A25. If greater accuracy is required, the temperature of the condensation may be taken and the heater credited with the difference in total heat of the steam entering the coil and the water leaving the coil. It is sometimes difficult, however, to maintain such constant conditions in the coil as will provide a constant temperature of the condensate, unless the steam pressure used for the test is materially above that suggested in the Code for low pressure heaters.

A26. Any error from this source produces a rating that is less than the actual, but that in amount is negligible for the purpose of the Code.

A27. The Code requires no correction for the barometric pressure or humidity in applying the factor 55.3. This error would influence the air capacity rating in either direction, depending on whether the barometer and humidity during test were high or low. For greater accuracy, use the factor correctly computed for the humidity and air density as of test.

A28. In addition to assuming a constant volume of air at the fans for a given fan speed, regardless of changes in the temperature of the air, the Code further assumes that the fan motor speed will not change when handling a given quantity of air at the varying densities of the air that will result from varying its temperature.

A29. This will not be true in practical operation where heaters are equipped with motors whose speeds vary with the load. This fact has no bearing on the Code test procedure and the methods of computations which deal only with fan speeds, without consideration of the size and characteristics of the motor necessary to produce those speeds and which result in true ratings for a given entering

air temperature, steam pressure and fan speed. For greater accuracy in rating, however, it would be necessary to determine the speed at which a given motor would drive the fans when handling air at the temperature which would result with a given condition and rate the heater at that speed.

DISCUSSION

L. W. CHILD: The code has been very thoroughly studied and it lines up almost exactly with our method of testing suspended unit heaters. However, we have added a condensate cooler to the condensate line. Sometimes we test with steam under pressure of 60 to 100 lb and without a condensate cooler we find the water bursts into steam and our percentage of loss is large.

For example, at 5 lb there is a possibility of 1½ per cent error due to not having condensate cooled down to room temperature. That is the maximum possible error.

Another point to be considered is the method of checking the amount of air. In our calculations we check the amount of air that the fan is delivering. We make this check with a Pitot tube or by means of a calorimetric method, using an electric heating coil and a watt meter. Temperatures are taken on each side of the coil, the thermometers being shielded from the radiant heat emitted by the electric element which otherwise would give quite an error.

One code requirement that we do not agree with specifies that the initial temperature is to be taken at some place which would give a reasonable degree of accuracy. We have yet to find a place in a suspended unit that will give a reasonable degree of accuracy, although we have tried every method to get them within 2 deg. We have even set the unit in rooms where we have had initial velocities through the room of 600 fpm and tried to obtain accurate temperature readings at the inlet, but we have not been able to do so. Therefore, I feel that we should take the inlet temperature as closely as possible and then having the air volume from the check method, and having the final temperature accurately, we can compute the initial temperature from these two figures and from the amount of condensate collected.

Then, if we find that the initial temperature that we get by computation corresponds to the average temperature that we have read over the inlet of the unit, we can be pretty certain we are right. In other words, we use inlet temperatures as a check instead of cubic feet per minute.

O. G. WENDEL (WRITTEN): The comments which I desire to make on the code will be taken up in three parts as follows:

1. The Code Edition January 1930.
2. Recommended revision of testing equipment.
3. Recommended revision of test computations.

The code edition January 1930 starts out with definitions. It is believed that the code should start with a statement regarding the unit heater as follows:

The unit heater tested should be a stock unit standard in every respect containing a steam tight heating element at the steam pressure tested. It is believed also that the unit heater fan and motor should be run for approximately a day before testing to break in the bearings.

Article 1 under *Definitions* defines standard air weighing 0.07488 lb per cubic

foot corresponding to dry air at 70 F or air at 50 per cent relative humidity with dry bulb to 68 F and barometric pressure of 29.92 in. mercury. The weight of dry air at 70 F and 29.92 in. mercury is generally considered at 0.07492 lb per cubic foot. If such is the case, it is the writer's opinion that the standard air should be limited to one condition.

Article 15 states that four tests must be run in order to obtain one set of rating data. It is believed that the number of tests necessary could be reduced to two tests. More will be said about this later under Recommended Revision of Testing Equipment.

Article 45 states that the test pressure should be controlled manually. It is believed that test pressure control could also be accomplished by an automatic pressure reducing valve.

Article 56 introduces a value of 55.2 in formula 2 which is incorrect if based on the values given in Article 1 for weight of air and specific heat of air. Using the values of Article 1 this figure would be 55.3.

Article 59 and Article A-26 state that the code is based on the assumption that for a given fan speed, a constant volume of air is handled by the fan or fans at the temperature in the fan or fans regardless of air temperature changes. This procedure is not followed out in the computation formula under Computation of Results, nor in the sample computations which include formulae and computations based on equal weight of air handled at a given fan speed regardless of temperature changes in the fan or fans. The code therefore is inconsistent.

The sample data show values for *temperature rise at 60 F entering 2 lb steam* and for *Btu with 60 F entering 2 lb steam* which are not correct if computed according to the prescribed formula. The values listed are consistent and approximately 2 per cent too high. It is not believed that Article 12 implies that 2 per cent should arbitrarily be added to computed results.

Article A-14 implies that thermocouples should not be used for air temperature measurements. It is believed that the last sentence of Article A-14 should be stricken from the code because the required accuracy is readily obtainable with a proper thermocouple installation.

Recommended Revision of Testing Equipment. The necessity of four tests required by the code is due to the uncertainty of duplicating free air discharge performance in the zero static chamber connected to the unit heater air outlet or outlets. It is believed that the introduction of the zero static chamber at the inlet side of the heater will reduce the testing requirements to two tests per set of output data.

The following brief description of the recommended test set-up and procedure can be used to accomplish these results:

1. Place the unit heater in an air tight zero static chamber permitting the unit heater to discharge freely outside of this chamber.

2. Measure the air supplied to the zero static chamber in a round duct with air flowing at 2,000 to 3,000 fpm by means of a Pitot-tube.

3. Compute outlet air temperature from condensation and air volume.

4. A second test should be run without the zero static chamber or with one or more walls of the zero static chamber removed so as to obtain the condition of free air delivery to and discharge from the unit heater. This test should ob-

tain condensation of quantity within plus or minus 2 per cent of the first test. It is assumed that both tests are run at same fan speed, same steam pressure and inlet air temperature. Where the latter varies from that of the first test, proper condensation corrections should be made to obtain the comparative condensation value.

The foregoing testing equipment is similar to that of the code except for the type and location of the zero static chamber.

Recommended Revision of Test Computations. The latter part of June 1928 I was called upon to compile certain unit heater performance data. In so doing it was necessary to evolve a method of computation based on equal volume of air handled at a given fan speed regardless of the air temperature in the fan or fans, using as a basis results obtained from tests.

This method of computation starts with the equation of the logarithmic heat transfer coefficient given in terms of the velocity of standard air, or volume of standard air if desired, passing through the unit heater. This information, determined from actual test data of three tests run at three separate fan speeds, is used together with the heating surface of the heater in setting up a performance equation, all factors of which are known but the final air temperature for any prescribed condition of volume of air handled, measured at the temperature of air in the fan, steam temperature and inlet air temperature. The final formula so obtained will take the form—

For the draw-through type of unit heater:

$$C = \left(Vt_f \times \frac{530}{460 + t_f} \right)^{1-n} \times \log_e \left(\frac{T_s - t_e}{T_s - t_f} \right)$$

For the blow-through type of unit heater:

$$C = \left(Vt_e \times \frac{530}{460 + t_e} \right)^{1-n} \times \log_e \left(\frac{T_s - t_e}{T_s - t_e} \right)$$

where

C = a constant for the heater $= \dfrac{C_1 S}{1.087\ A}$, in which

C_1 = a constant of the equation, $K = C_1 V^n$ (std), and

$\dfrac{S}{A}$ = heating surface per unit free area of heater

Vt_f = velocity of air at t_f through heater free area feet per minute

Vt_e = velocity of air at t_e through heater free area, feet per minute

t_e = temperature of entering air

t_f = temperature of final air

T_s = temperature of steam

n = exponent of velocity in the coefficient equation

The foregoing method of computation presents a basis for the determination of output data for steam pressures and inlet air temperatures other than that required for standard 2 lb steam and 60 F inlet air.

Since unit heaters are invariably selected for inlet air temperatures and steam pressures, other than the conditions of standard rating, 60-F inlet air

and 2-lb steam, it is believed that provision should be made in the Code to cover such output data.

MR. CHILD: Mr. Wendell has brought up the fact that in a unit heater the *K* value against air velocity has a definite mathematical relation. On draw-through units I have found by test this seems to be the case when pounds of air per minute are used as velocity, but where we have a blow-through unit heater, with a disc fan in back of it, I have yet to be able to get any pure mathematical relation between *K* value and mass velocity of air.

On the suspended type of unit heater, that is the blow-through type, due to the fact that there is a great deal of eddying caused by the fan, the *K* value of the unit for a given mass velocity is always greater than the *K* value found when the unit is placed in a duct and with a separate fan in place of the disc fan. This phenomenon does not seem to be present in the draw-through units. Therefore, I have found that we cannot take two or three points and plot a curve on logarithmic paper of *K* versus mass velocity and get a straight line. We find it necessary to plot the curve on plain coordinate paper so as to determine the *K* value for different fan speeds. The fact that some disc fans have various characteristics at various speeds causes this varying *K*-value curve. Incidentally *K* values should always be predicted on the logarithmic mean temperature difference between air and steam.

I have taken test data and computed results by the test code method and by the theoretically correct method and find a difference of only 1.4 per cent in the results. This difference was due to the large temperature difference between air and steam. I therefore feel the test code is entirely satisfactory for all normal work.

E. J. VERMERE (WRITTEN): Where mechanical circulation of air is employed in connection with the distribution or diffusion of heat, we know that a measurement of the weight of steam condensed is not by itself a criterion of effective heating results. This theory has in a limited way been very clearly demonstrated by the results obtained in the series of tests at the University of Illinois on radiators with and without enclosures. Professor Willard and associates gave a summary of these results in the paper entitled, Investigation of Heating Rooms with Direct Steam Radiators Equipped with Enclosures and Shields, TRANSACTIONS, 1929.

A summary of this test work clearly shows that more effective heat distribution or diffusion can be obtained by directing the gravity flow of air in a definite manner. It is also shown that the amount of heat or rather the amount of condensation is reduced approximately 15 per cent with the ideal application. In general practice we have proportioned the amount of heating surface solely on its condensation rating. Therefore, for an application exactly as recommended, we would increase the amount of heating surface from 10 per cent to 25 per cent. We can now understand why the system may be entirely out of balance. Are we not starting our unit heater test work in exactly the same manner?

The proposed code as far as the mechanics of obtaining accurate condensation, flow of air, and temperature rise data, is very exacting, and all of this information must be obtained by some uniform means. According to our analysis the total of all the percentages of error which may be obtained under this method of testing is approximately 2.6 per cent.

As far as effective heat distribution is concerned, this proposed code entirely ignores heat diffusion. Is it not possible to establish an additional factor based on outlet air temperature, volume of air circulated, and height of discharge above floor level? By applying this correction factor to the heat output of the unit, the engineer can always be assured of the most successful heating results. When we ignore both the velocity and the temperature of the air leaving the unit, we are then establishing a rating solely based on condensation.

We realize that two unit heaters of greatly different air volumes and outlet temperatures, may both put into the space to be heated the same number of heat units per hour. The low velocity unit may recirculate 4000 cfm with an outlet temperature of 160 F and the high velocity unit 6700 cfm with an outlet temperature of 120 F.

The heated air leaving the unit is pulled towards the upper areas of the room by the aspirating force. This force of buoyancy is proportional to the temperature differential between the air leaving the heater and the room temperature. The only force operating against this aspirating pull is the force or velocity pressure of the moving stream of air. The greater the temperature differential, the greater the aspirating pull, and the lower the velocity the less the velocity pressure.

We know that outlet air temperature is important in connection with unit ventilation work where good diffusion of the air within the room must be obtained in order to give good ventilation results. In order to prevent air stratification which in turn prevents proper diffusion, outlet air temperatures are limited in this type of work to between 100 and 110 F which of course depends upon the outlet velocity. This fundamental principle of diffusion must certainly apply to unit heater application.

The question may arise as to why this fact has not been self evident long ago. In designing unit heater applications, we figure the heat losses from the building exactly the same as we would for an installation of direct radiation. In selecting the number and size of unit heaters on the basis of a condensing capacity equal to that required with direct radiation, we then have sufficient capacity to heat the space from the top down exactly as occurs with the convection heating of direct radiators or pipe coils. What should the owner pay for this excessive heating capacity?

The greatest service our Society can do is to start at once to obtain data regarding heat diffusion in relation to temperature differential and outlet velocity. Professor Willard has clearly shown what may be expected along these lines. This is altogether too big a subject for a manufacturer to tackle by himself, and in order to insure reliable information along these lines, this work should be sponsored by our Society. For the present the only method of rating unit heaters must necessarily be that of steam condensed per hour.

W. A. ROWE: There has been a necessity for a code without a question of doubt. Since the advent of fin-tube radiation and cellular type radiation the unit heater has made tremendous progress. I was told last week that there were 63 manufacturers of unit heaters; and that was last week. There has been a necessity for a measuring stick and we have been working for two years to get something that everybody can use. Our Committee was composed partially of fan men and partially of men who have not been active in fan work. As

pointed out at the Bigwin meeting in June, 1930, there were differences of opinion in the group. Nobody questioned the method of getting the condensation, and I do not think that needs any particular comment. The only question was, How to obtain the correct volume of air? It was desired by the fan men that they measure the air, but when we tried to put a Pitot tube test in the code some of our friends were not quite so consistent in the results they got; they could not get them very close.

It was suggested that the volume be calculated from the temperature rise and condensation. The best advice we could get was that you could not get a result with that method correct within probably better than a difference of the order of 4.5 per cent. Well, we then called on the Society for help. The University of Kentucky was asked to tell us how close they could come to the correct volume of air by the two methods. Professor O'Bannon undertook those tests which were reported at the last meeting. As I recall it, his average temperature variation was 2.5 per cent with thermometers, and with thermocouples 0.5 per cent. Exceptions have been taken to thermocouples. Resistance thermometers were preferred. Your Code Committee very wisely left out that debatable point and let you use what you please as long as you take the correct temperature, but it satisfied us that either way is all right, and if that is true within reasonable tolerance, we are going to pick the easy way.

The easy way is by calculation from the measurement of the temperature rise, if it is surrounded with reasonable precautions. The first is to mix the air and, secondly, measure its temperature at a reasonable high velocity. This we have done. Finally the question arose, Shall we check the volume by direct air measurement? We did not say at first that it had to be done. Some of the more technical men said, That is not right, somebody is going to make a poor heater. If the heater should not free itself readily of condensate you are not going to get the right amount of water. You should check the air volume to be sure.

Now, measuring the temperature at that high velocity in the nozzle lends itself beautifully to a nozzle test, and anybody who has tested for air volume knows how simple it is with a calibrated nozzle to get the air velocity with one reading on the manometer and how much more accurate it is than a Pitot tube traverse. And so we have a very happy solution. You do not have to have a test duct 10 or 15 diameters long. All you need is a pipe to attach to a fan inlet and draw through that orifice and get the velocity reading. At the same time you get the temperature rise.

There is one more point. We have had very able engineering advice on this code. Some of us have not been engaged in the actual test work for many years. Some of us have forgotten most we ever knew about the highly scientific side of it, but we have had some good brains on the committee. Our worthy new president was with us and Professor O'Bannon and members of the engineering force of the different groups. We know that there are a lot of hair-splitting differences that you can not account for. We did not quibble about how much heat the motor gave to the air current because we were going to lose a little heat radiated from the housing. We did not worry whether there was 10 deg of superheat or 2, as long as there was some.

We have a code that is easy to use; it is accurate within a range of a commercial exactness and that is what we are after, and that is what we need, and

if everybody uses it we are all in the same boat. The yardstick is not 39 in. long; it is pretty close to 3 ft. I think we can spend too much of this meeting's time to question little changes. If there are corrections to be required in the mathematics, they will be taken care of in the new draft, because it has to be rewritten to take out some of the typographical errors and to provide for the change recommended in making the air volume test mandatory. I would like to see the Society adopt this code, because it is all right and it will afford an opportunity to take up some other important work, but right now we want this code and we do not want to let it go another year.

L. S. O'BANNON: I just want to answer one point that the speaker brought out and that is that the method described in the code for correcting for entering air temperature and steam temperature is certainly based on the principle of constant volume at the fans. In spite of the fact that during some of the computations constant weight is assumed, later on in the method corrections are made in a logical way for the original assumption of constant weight to bring the calculations in agreement with the method of constant volume.

D. E. FRENCH: The condensate cooler suggested by Mr. Child is recognized in the Addenda as a provision for greater accuracy than is thought necessary for the purpose of the Code. A check of the air quantity determination by alternate methods is highly desirable. The Code Committee chose a calibrated nozzle for a check by direct measurement, as being perhaps the simplest to use.

Mr. Child is correct in his observation that it is difficult to hold the entering air temperature constant over the plane of the heater inlet and to read a true average. Here, however, the variations can be held to 2 deg if the air is drawn from a room that has liberal openings to the outside and the output of the unit being tested is delivered outside. Then if a number of thermometers are disposed over the inlet with the judgment that may be expected of a test engineer, their average should be true within the accuracy contemplated by the Code. Mr. Child's suggestion that the inlet temperatures be checked back from the outlet temperatures, assumes that the reading of the latter is more accurate. This is a false assumption. We have found that in spite of all the precautions of the mixing chamber and a high velocity area where the temperatures are read, the variation at this point is also in the order of 2 deg. However, this represents a great improvement by comparison with the variation of 30 deg found over the face of some disc fan heaters and 10 deg over the outlets of multi-housed fan heaters, when the mixing chamber with high velocity outlet is not used.

Mr. Wendell's discussion which I had not seen prior to its reading at the meeting, contained more subject matter than could be properly assimilated and discussed on the floor of the meeting. Thus I want to present the following answer to his discussion after having an opportunity to consider it with the care that Mr. Wendell's interest in the Code deserves.

It is true that the Code will be most used by manufacturers for the test and rating of standard units, but to confine it to use with stock units, as Mr. Wendell suggests, would restrict its usefulness for the test and rating of units especially designed for individual applications, of which so many are produced.

It would be in the direction of further accuracy to require that the unit be operated for a day before test, to break in the bearings, but this is not necessary

in the opinion of the Committee, because excess bearing friction in the amount that it would be necessary to consider, is extremely unlikely in present day manufacture and further because the Code is primarily for the benefit of the purchaser, and the manufacturer can be assumed to avoid anything that would result in a too high power rating.

Standard Air is defined as shown in the Code, in order to agree with the Standard Code for the Testing of Disc and Centrifugal Fans, which has already been adopted by this Society.

While the steam pressure might be controlled automatically, the equipment therefore could not but add materially to the expense of the test apparatus and would not, in the opinion of the Committee, eliminate the need for constant observation of the manometer as a check. Thus it seemed better in the interest of simplicity and economy to depend upon manual control.

Mr. Wendell's observation that the value 55.2 should be 55.3 for the weight and specific heat of air given in the Code is correct. This change will be made.

Professor O'Bannon's remarks and a further study of the Code will have shown Mr. Wendell his error in believing that the Code is not consistent with the assumption of constant volume at the fans. Undoubtedly he has confused weight of air and volume of air. The volume of air at the temperature of the fans is determined by means of the first two formulae for the fan speed of test. This volume remains constant for that fan speed, regardless of conditions of entering air temperature and steam pressure.

The succeeding formulae and graphical solutions are for the sole purpose of determining the heat output for any other condition of entering air temperature and steam pressure than those of test and have nothing to do with air quantity rating. This output for some other condition of entering air and steam pressure might be determined directly from the output of test in the proportion of temperature difference between entering air temperatures and steam temperatures, if it were not for the fact that this proportion applies only where the mass velocity through the coil remains constant.

It is obvious that if the air entering the heater were at some other temperature than that of test and the volume remains constant at the fans, the weight of air passing through the heater and therefore the mass velocity through the coil would not be the same as on test.

In recognition of this fact, the code next determines at what speed the heater would have to run to handle the same weight of air as of test at the desired entering air temperature and steam pressure, and what the heat output would be under these conditions. On the curve plotted with these values for each of the three test speeds, the output of the heater can be read for the standard fan speed.

The foregoing will also clear up Mr. Wendell's criticism of the values computed for temperature rise with 60 F entering, 2 lb steam, and Btu with 60 F entering 2 lb steam. Naturally it is not the intention of article 12 to imply that the tolerance factor may be added arbitrarily to obtain a rating higher than that justified by test.

Mr. Wendell is right in suggesting that the last sentence of Article A-14

implies too strong a criticism of thermocouples for this use. This sentence will be omitted.

As a preface to a discussion of the alternate methods of test and computation recommended by Mr. Wendell, let me make it clear that there are many methods other than those prescribed by the code which may be equally accurate and that have much to recommend them. It was the task of the Committee to consider as many of these as it could get information on and select one which had the most to recommend it for the purpose of the code and for which could be obtained the endorsement of the majority of the engineers and manufacturers who specialize in unit heaters.

The test set-up requiring a zero static chamber at the heater inlet, as suggested by Mr. Wendell, is one of those considered by the Committee. The main objection was the fact that this method depends entirely on direct measurement of air quantity and does not allow a check of this measurement by condensation temperature rise determination.

Of the four check tests required by the code for calibration of the receiving chamber at the heater outlets, Mr. Wendell's method would eliminate but three of ½ hour duration each, during which it is necessary to measure only the condensation and chamber pressure. These, therefore are not laborious and their elimination is not nearly so much to be desired as the value of having a check of the air quantity.

The Committee's chief criticism of the method proposed by Mr. Wendell for computation, is the same as that expressed by Mr. Child. Mr. Wendell's method depends upon the coefficient *K* separately determined for the heating surface. He assumes that this coefficient varies only with velocity and temperature difference. This is not the case with disc fan, blow-through heaters for instance, where the coefficient depends also on turbulence which, with the same quantity of air going through the heater, may vary with the type of fan, the distance of the fan from the surface and the extent to which the diameter of the fan covers the area of the coil.

Mr. Wendell has said that the code should have provision for determining ratings at alternate entering air temperatures and steam pressures. He has overlooked the fact that the code has such provision. As stated in Article 58, a standard procedure is here given whereby data of test may be used for the determination of performance under *any* desired conditions of rating. The code is illustrated with sample computations and curves which convert the test results to the basis of 2 lb, 60 F. However, values for any other entering air and steam conditions can be determined by substitution of the corresponding temperatures.

It is true that this method is not available to the engineer in the field who may want to convert from a rating at one set of conditions to that at another set of conditions which may not be shown in the manufacturer's published data. However, the Committee has been unable to devise an accurate method which is independent of the characteristics of the heater and could therefore be applied equally well to all types of heaters.

So far, we are sure only of accuracy where the method is based on data which reflects the characteristics of the individual heater.

Until such a method is found, it will be incumbent upon manufacturers to rate at a sufficient number of entering air temperatures and steam pressures to cover

every range of use, depending upon the engineer to interpolate for approximate values at intermediate conditions not shown.

An alternate method in use by one manufacturer is to devise from test data reflecting the characteristics of the heater in question a table of factors by which the heat output rating of that heater can be multiplied, to determine the rating for any condition of entering air and steam pressure.

Mr. Vermere's plea for a code that will rate unit heaters in terms of effective distribution voices a creditable ambition that this Society might well entertain, but not without a clear realization of difficulties that Mr. Vermere's paper fails to express. I feel sure that in reading this discussion as bearing on the code, it was not so much Mr. Vermere's intention to criticize the code, as it was to use the occasion for drawing attention to a need beyond the present scope of the code. Indeed the code is best construed as the first step and the only step just now practicable towards the end he seeks.

When the code is adopted, published and generally accepted, the public will have uniform ratings that are comparable and a rating will include all of the information necessary to an intelligent selection. This is the first step. Thus the greatest need is taken care of. Next, it might be desirable to guide the public with some evaluation of the factors making for economy in different applications, but it must be recognized that progress in this direction must be slow and laborious.

It has taken three years for a representative majority of test engineers and manufacturers most familiar with unit heaters to agree on a relatively simple problem of standardizing methods of physical measurement under conditions that are easy to define and make uniform—methods prescribed to evaluate characteristics that are common to all products of the class without respect to design or application and that bear no reflection on the engineering judgment of the designer.

So, it will be a laborious proceeding to establish agreement on such a complex subject as the distributing effect of unit heaters, which depends not alone on attributes of the unit itself but on surrounding conditions, purpose of use, characteristics of buildings and relative location of the unit. It will require the reconciling of widely opposed engineering views that now obtain in this stage of development of a relatively young industry.

I am heartily in favor of the Society's embracing such a project, but with full realization of the following conditions:

1. Unit heaters have many uses requiring different characteristics for effectiveness or economy. Thus a *code for unit heaters* would not do. There must be a separate *code* or manual for the heater in application to each broad use. Taking for example only those factors dwelt on by Mr. Vermere, a temperature rise most effective for recirculating heating would not be sufficient for a ventilating unit and that best suited to ventilating might still be insufficient for a drying purpose. An outlet velocity suited to a large area might be intolerable for a smaller room. A location effective for room heating might not be best for vapor absorption or removal of roof condensate.

2. A single committee can only simplify and eliminate to some degree the need for the application of engineering knowledge.

3. Nor can a single committee assume authority to select among the prin-

ciples that are still contested among specialists of wide resources and experience, where these principles cannot be reduced to positive test and remain a matter of opinion until agreement is reached through growing experience.

4. The labor of collecting data and the endless tests required to determine the value of any one of the factors to be rated, with that degree of accuracy expected of a code in the usual sense, would be entirely beyond the resources of the Society.

Growing experience in the industry has brought general agreement on some principles. The value of a high air ratio pointed out by Mr. Vermere is an example. These could be valued and standardized within reasonable limits for certain broad uses and widely published to make general acceptance certain. Agreement on other principles may only await the cataloging of opinion and data already available, by a committee of broad view, such as the Society might provide.

Thus it might be practicable for a Society committee to undertake the work of assembling available data, with the intent to provide a manual of selection and application of unit heaters which might insure general recognition of accepted standards and guide engineers in their selection for a given purpose, from the range of unit heaters available.

Such a committee would find an ally in the Engineering Committee of the *Industrial Unit Heater Association.* That Committee is now discussing the problem of accelerating that standardization that would come in time in this industry as it has in others, through the gradual elimination of opposing designs as growing experience dictates the better practices.

PRESIDENT LEWIS: Are you ready for the question? The motion before us, made by W. A. Rowe and seconded by Perry West, provides that the Report of this Committee as presented be adopted as the Society's Standard Code for Testing and Rating Steam Unit Heaters with such minor corrections as may be approved by the Committee before the final printing of the Code and its distribution to members of the Society. The motion was carried.

No. 856

SUGGESTED METHOD OF TESTING UNIT HEATERS SUITABLE FOR FIELD USE

By L. S. O'Bannon,[1] Lexington, Ky.

MEMBER

IN TESTING unit heaters it has been the experience of most observers that the warm air leaving the unit does not have a uniform temperature or velocity. The average of the readings of several thermometers located at the discharge outlets *may not* give the true average temperature of the total weight of air discharged. Instances have been reported, however, where satisfactory temperature readings have been made in this way but these instances seem to be exceptions. Accuracy in observing the final temperature of the air is vital to the condensate-temperature rise method of computing air volume. If this method is used it is necessary either to obtain a weighted average, which requires the determination of velocities, or to force a uniform temperature by providing some means of mixing the air. The latter system is generally favored. The weighted average method is seldom attempted.

It is the purpose of the writer to describe a method by which the air volume output may be obtained accurately with a minimum amount of auxiliary equipment. The method may be used in the laboratory or in the field. It is especially recommended for field tests, first, because of its simplicity, and second, because its simplicity is not secured by sacrificing accuracy.

The method has been named the condensate-nozzle method. The underlying principles are the same as those upon which the condensate-temperature rise method is based. The heat output of the unit is found in the usual manner by measuring the steam condensing capacity of the heater. Air volume outputs are computed indirectly from the temperature rise and the heat given up by the steam. The distinctive feature of the condensate-nozzle method is the graphical means employed to arrive at certain unobserved final temperatures. The procedure is as follows:

Let it be required to obtain for three different fan speeds, say 1750, 1160 and 870 rpm, the heat output, air volume output and final air temperature of a unit heater of the draw-through type illustrated in Fig. 1. The steam pressure is to be 5 lb per square inch. The unit is arranged for a heat output test

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as shown, and, it should be noted, is free from any abnormal restrictions to air flow.

Three heat output tests are made, as near the specified speeds as possible. Let it be assumed that these tests yield the data as given in Table 1, Section A. No observations of final temperature were made and therefore it is impossible to calculate the air volume output.

The next step is to remove the diffusing outlets of the unit and place in their stead a venturi-shaped nozzle as shown in Fig. 2. The inlet section of the nozzle covers all the outlets of the unit. The throat has a relatively small diameter sufficient to cause an air velocity of 1,000 fpm or more as determined

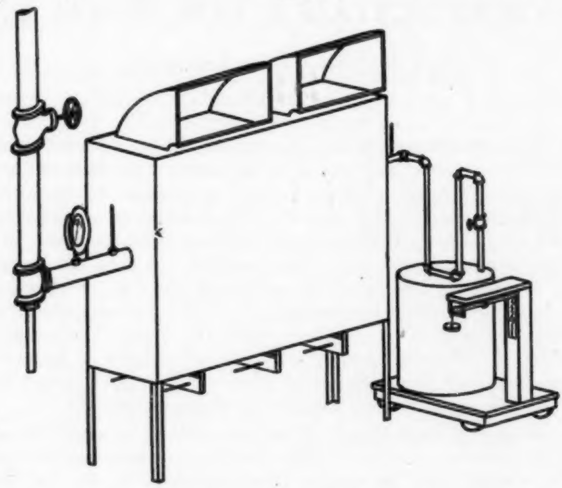

FIG. 1. UNIT HEATER ARRANGED FOR HEAT OUTPUT TEST WITHOUT ANY ABNORMAL RESTRICTIONS TO AIR FLOW

from the lowest air volume output anticipated. The purpose of the diverging outlet cone is to reduce the over-all resistance of the nozzle. Four thermometers are inserted at the throat section for measuring the final temperature. Three more tests are now made with results as shown in Table 1, Section B.

If the data of tests, 4, 5 and 6 are used to plot a curve showing the relation between final temperature and heat output this curve can be referred to, to obtain the final temperatures for tests 1, 2 and 3. However, the relation between heat output and final temperature varies with the entering air temperature. Therefore, it is necessary to reduce the data of tests 4, 5 and 6 to the same entering air temperature as tests 1, 2 and 3. This requires three separate transformations since the values of the entering air temperature for the first three tests are not the same. Leaving the method by which the transformations are made for later discussion, Table 1, Section C, gives the corrected values.

The curves are plotted as shown in Fig. 3. The final temperatures corresponding to the heat outputs of tests 1, 2 and 3 are read from the curves, thus completing the data of these tests as shown in Table 1, Section D. Since an entering air temperature of 60 F is used as a basic temperature for the purpose of rating, the transformation to this base may be made most con-

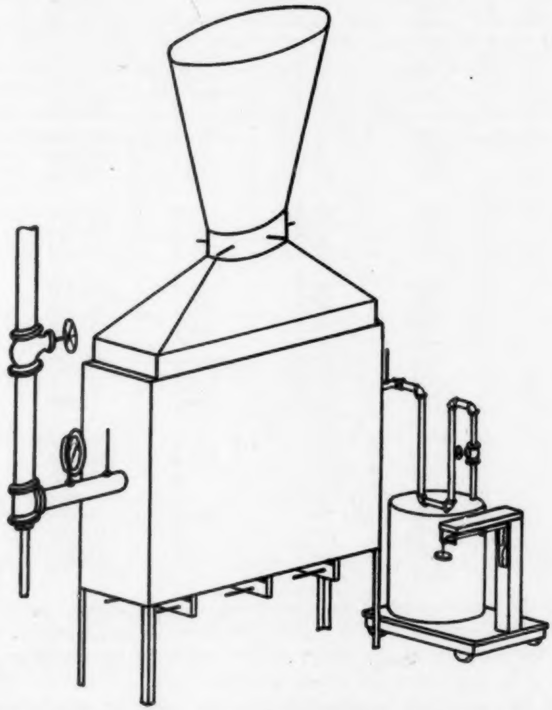

FIG. 2. UNIT HEATER WITH VENTURI NOZZLE ATTACHMENT FOR AIR VOLUME TEST

veniently at this stage before continuing the calculations. The converted figures are given in Table 1, Section E.

The air volume outputs are now calculated. The curves are plotted as shown in Fig. 4, giving the relations between fan speed, the volume of air in cubic feet per minute, the final temperature and the heat output in Btu per hour. It only remains to select from these curves the proper values corresponding to the three fan speeds specified at the beginning. The results are summarized in Table 1, Section F.

If it had been required to find the output of the unit at only one speed, foı

TABLE 1. SUMMARY OF TEST DATA

Section	Test No.	Fan Speed rpm	Inlet Air Temp.—F	Outlet Air Temp.—F	Heat Output Btu Per Hour	Air Volume cu ft per Min at Fans
A	1	1751	87.9	.....	182,900	....
	2	1184	87.8	.....	135,500	....
	3	788	90.7	.....	97,600	....
B	4	1752	68.9	133.0	205,300	....
	5	1175	69.7	139.8	153,400	....
	6	794	74.1	149.2	109,600	....
C	4	1752	87.9	145.1	179,600	....
	5	1175	87.9	150.6	134,800	....
	6	794	87.9	156.7	99,200	....
	4	1752	87.8	145.0	179,600	....
	5	1175	87.8	150.5	134,800	....
	6	794	87.8	156.6	99,200	....
	4	1752	90.7	146.8	175,600	....
	5	1175	90.7	152.3	132,000	....
	6	794	90.7	158.3	97,200	....
D	1	1751	87.9	144.9	182,900	....
	2	1184	87.8	150.5	135,500	....
	3	788	90.7	158.2	97,600	....
E		1751	60	126.8	221,000	3403
		1184	60	133.8	164,000	2310
		788	60	141.3	120,900	1567
F		1750	60	126.8	221,000	3390
		1160	60	134.2	162,000	2280
		870	60	139.7	130,000	1720

example, at the rated speed of an alternating current motor, the three points required for the curve in Fig. 3 could have been obtained just as readily by dampering the air flow at the outlet of the nozzle or by inserting orifices at the throat.

Although the diverging cone of the nozzle helps to reduce the over-all resistance and minimizes the hazard of extrapolating to get the final temperature at the maximum output, this cone is not essential. If there is some means available for increasing the normal speed of the fan, it is possible to exceed the normal maximum output, eliminating extrapolation entirely and permitting the use of a nozzle of higher resistance. A substitute motor of higher speed may be used; a d-c motor may be run for the short period of the test at a few hundred revolutions above its normal speed; an auxiliary motor with belt drive may be used.

It should be noted especially that this method is not dependent upon the accurate reproduction of free delivery conditions within the confines of the duct placed over the outlets. The method does not prohibit the use of an auxiliary fan to overcome the resistance of the collecting nozzle or to exhaust the heated air out of the test room, but the method emphasizes the fact that

the auxiliary fan is not essential from the standpoint of totally equalizing or balancing the resistance of the nozzle; furthermore, the method is not concerned with the volumetric proportions of the nozzle, or other mixing device which may be substituted for it.

Attention has been called to the suitability of this method for field tests. While there may not be much demand for field tests of unit heaters there may be installations where ventilation is of prime importance; and outside air connections, mixing dampers and filters may impose frictional resistance materially affecting the air volume. In such instances it may be desirable to have a relatively simple and inexpensive method by which the air volume output may be accurately determined. Even in cases of this kind, a field test designed to determine air volume especially is not necessary if the unit in question has been tested according to the rules of the *Industrial Unit Heater*

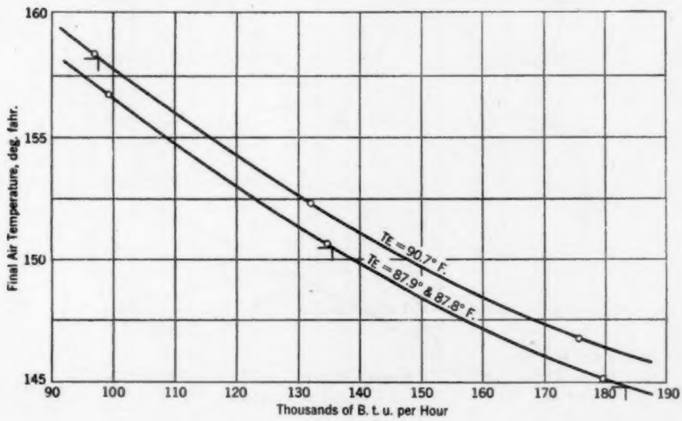

FIG. 3. RELATION BETWEEN FINAL AIR TEMPERATURE AND HEAT OUTPUT

Association. Because, as the foregoing discussion bears out, it is only necessary to measure the steam condensing capacity of the unit and the entering air temperature in order to obtain the air volume if there is already available reliable information in the form of curves or catalog data showing the relation between air volume and heat output. The majority of catalogs furnish these data. That is to say, if a unit heater is operating under abnormal restrictions to air flow, either at inlet or at outlet or both, the air volume output may be obtained by measuring the condensation rate of the unit and referring the heat output to a curve derived from the manufacturer's catalog data.

It would seem, therefore, of little use to have a method simple enough and inexpensive enough to take to the field. On the other hand, the availability of such a method may be welcomed by contractors, purchasers and consulting engineers. The method is also suitable for the occasional testing which may be undertaken in private laboratories or institutions where the extent of the work would not justify more elaborate equipment.

Contrasting this method with the direct measurement of air volume by Pitot tube or other flow meter and with the method of artificial reproduction of free delivery, the advantages are:

1. Small amount of test equipment required.
2. Minimum amount of test labor required.
3. Minimum floor space required.
4. Relatively inexpensive, quickly set up, and readily adapted to the occasional test in laboratory or field as contrasted with the more elaborate and permanent set-up probably more convenient for rapid routine testing.

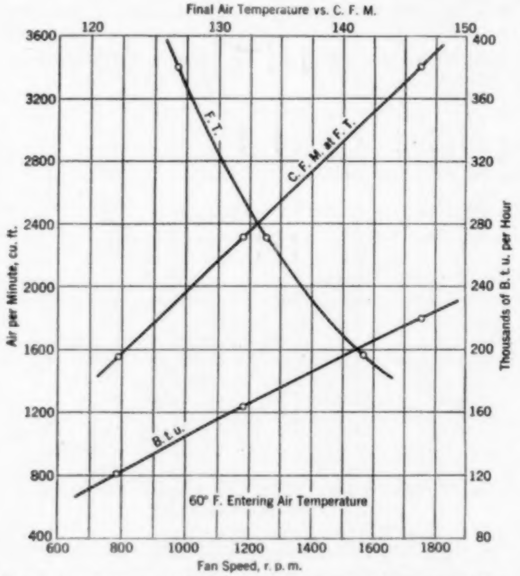

FIG. 4. CURVES SHOWING RELATION BETWEEN FAN SPEED, HEAT OUTPUT, AIR VOLUME AND FINAL TEMPERATURE

5. The air volume output, although obtained indirectly, is the volume produced without abnormal restrictions to air flow.

The disadvantages are:

1. No single test is complete in itself.
2. At least three extra tests are required regardless of the number of fan speeds specified for which data are wanted. This is time-consuming and is a serious objection where a large amount of routine testing is performed.
3. Except under favorable circumstances, inability to get rid of heated air in the test room.
4. Unless special attention is given to their design, individual nozzles are not adaptable to a range of unit heater sizes or shapes.

Fig. 5 illustrates other applications of the nozzle. On account of the rela-

tively small surface area of the nozzle between the heater and the throat section, insulation is not very important, but the loss should not be disregarded and plenty of insulation should be provided.

In the preceding explanation of the test method, the manner of converting the test data from one entering air temperature to another was not discussed. Several investigators have shown that the air volume at the fan of a unit heater

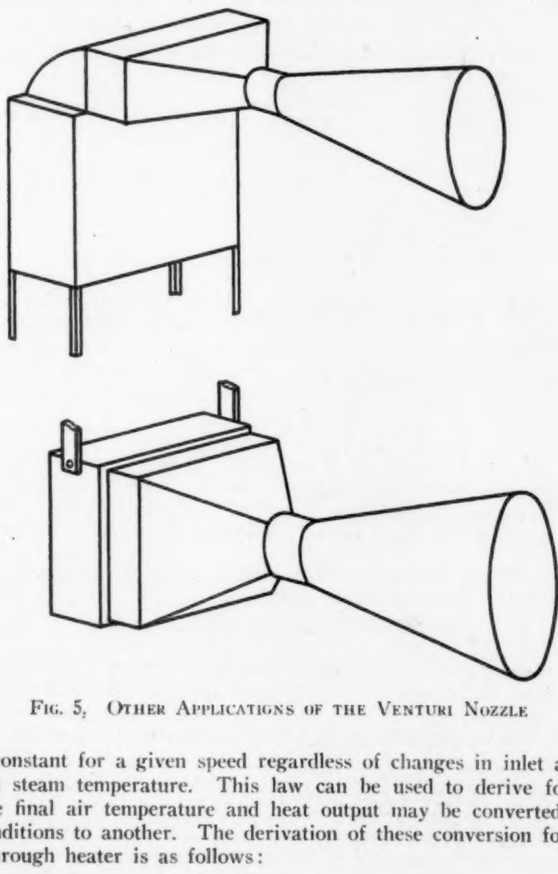

FIG. 5. OTHER APPLICATIONS OF THE VENTURI NOZZLE

remains constant for a given speed regardless of changes in inlet air temperature and steam temperature. This law can be used to derive formulae by which the final air temperature and heat output may be converted from one set of conditions to another. The derivation of these conversion formulae for a draw-through heater is as follows:

Let Q = Heat output, Btu per hour
W = Weight of air, pounds per hour
V_F = Volume of air at fan, cubic feet per min
C_p = Specific heat of air
U = Coefficient of heat transmission

S = Heating surface
T_S = Steam temperature, degrees Fahrenheit, absolute
T_F = Final temperature of air, degrees Fahrenheit, absolute
T_E = Entering temperature of air, degrees Fahrenheit absolute
d = Density of air

Subscript—*1* refers to some known initial condition.
Subscript—*2* refers to a second condition to which the first is to be converted.
Then:

$$\frac{Q_1}{Q_2} = \frac{W_1\ C_p\ (T_{F_1} - T_{E_1})}{W_2\ C_p\ (T_{F_2} - T_{E_2})} \tag{1}$$

$$\frac{W_1}{W_2} = \frac{60\ V_{F_1}\ d_1}{60\ V_{F_2}\ d_2} \tag{2}$$

$$\frac{d_1}{d_2} = \frac{T_{F_2}}{T_{F_1}} \tag{3}$$

Combining (2) and (3) with (1):

$$\frac{Q_1}{Q_2} = \frac{V_{F_1}\ T_{F_2}\ (T_{F_1} - T_{E_1})}{V_{F_2}\ T_{F_1}\ (T_{F_2} - T_{E_2})} \tag{4}$$

But the volume at the fan remains constant.

Therefore $V_{F_1} = V_{F_2}$, and

$$\frac{Q_1}{Q_2} = \frac{T_{F_2}\ (T_{F_1} - T_{E_1})}{T_{F_1}\ (T_{F_2} - T_{E_2})} \tag{5}$$

or

$$\frac{Q_1}{Q_2} = \frac{1 - \dfrac{T_{E_1}}{T_{F_1}}}{1 - \dfrac{T_{E_2}}{T_{F_2}}} \tag{6}$$

Also, from the familiar heat transmission formula, $Q = US\theta_m$, in which θ_m is the logarithmic mean temperature difference:

$$\frac{Q_1}{Q_2} = \frac{U_1 S\ (T_{F_1} - T_{E_1})}{\log_e\left(\dfrac{T_{S_1} - T_{E_1}}{T_{S_1} - T_{F_1}}\right)} \div \frac{U_2 S\ (T_{F_2} - T_{E_2})}{\log_e\left(\dfrac{T_{S_2} - T_{E_2}}{T_{S_2} - T_{F_2}}\right)} \tag{7}$$

Assuming $U_1 = U_2$ and equating (7) to (5) we have

$$\frac{T_{F_2}}{T_{F_1}} = \frac{\log_e\left(\dfrac{T_{S_2} - T_{E_2}}{T_{S_2} - T_{F_2}}\right)}{\log_e\left(\dfrac{T_{S_1} - T_{E_1}}{T_{S_1} - T_{F_1}}\right)} \tag{8}$$

Rearranging equation 8:

$$\frac{T_{F_1}}{\log_e\left(\dfrac{T_{S_1} - T_{E_1}}{T_{S_1} - T_{F_1}}\right)} = \frac{T_{F_2}}{\log_e\left(\dfrac{T_{S_2} - T_{E_2}}{T_{S_2} - T_{F_2}}\right)} \tag{9}$$

It is apparent, therefore, that for any given heater and fan speed:

$$\frac{T_F}{\log_e\left(\dfrac{T_S-T_E}{T_S-T_F}\right)} \text{ is a constant.} \tag{10}$$

Making use of this relation, charts or tables can be constructed by means of which the new final temperature may be found corresponding to any new entering air temperature or steam temperature. After finding the new final temperature, equation (6) can be used to find the new heat output.

Similarly, for a blow-through unit heater, the following relation is obtained:

$$\frac{Q_1}{Q_2}=\frac{\dfrac{T_{F_1}}{T_{E_1}}-1}{\dfrac{T_{F_2}}{T_{E_2}}-1} \tag{11}$$

and

$$\frac{T_E}{\log_e\dfrac{(T_S-T_E)}{(T_S-T_F)}}=\text{a constant.} \tag{12}$$

The transformations involved in the example of test calculations described in the foregoing were made by means of the function just derived for draw-through heaters. It would have been simpler but not quite correct if the conversions had been made according to the assumption of constant mass velocity using the familiar relations

$$\frac{Q_1}{Q_2}=\frac{T_{F_1}-T_{E_1}}{T_{F_2}-T_{E_2}}=\frac{T_{S_1}-T_{E_1}}{T_{S_2}-T_{E_2}} \tag{13}$$

Using the constant mass formula, all of the tests could have been converted immediately to equivalent conditions at an entering temperature of 60 F and the intermediate steps made necessary by the constant volume method would have been unnecessary.

DISCUSSION

D. E. FRENCH: A great deal of credit is due Professor O'Bannon for his work which, I believe, was an outgrowth of the studies he made to help the Joint Code Committee. There is a distinct need for a simplified test method that can determine air quantity with reasonable accuracy and that involves simple equipment that can be readily applied in the field. We know that Professor O'Bannon was working on this method but until this time he has not been able to assure us that the method would apply equally well to all types of heaters, including the disc fan type, when submitted to the substantial resistance which his method imposes. The test that he has shown seems to indicate that he can prove that, and when he does the method would look readily applicable to the field test that we have been wanting.

In the case of the computations required to convert from one condition of rating to another, the code method involves a graphical procedure which must reflect the characteristics of the individual heater. If Professor O'Bannon

can go further with this method and produce a means by which the engineer in the field can take a manufacturer's catalog data at one condition of rating and determine an accurate rating at any other condition of rating, that would be an addition to our equipment that is very much to be desired.

I hope that some committee will be able to inherit and follow the work that Professor O'Bannon is doing so that we can bring something of that sort before the Society.

L. W. CHILD: The computations of Professor O'Bannon are very interesting. I have been using a method for the last several years based on the combination of the heat transfer formula and the fact that the heat is equal to the weight of air multiplied by the specific heat multiplied by the temperature rise. With a heating element whose temperature can be considered constant, such as we have in any steam heating element, a combination of these two formulae can be worked out to give a formula in which the final air temperature is a function of steam temperature, inlet air temperature, and a certain constant C. This constant is a function of the logarithmic base e raised to a certain power: said power being a function of the K value of the surface, and the mass velocity of the air passing over the surface.

Since the K value is also a function of the mass velocity, a curve may be plotted for any given surface; plotting the constant C against the mass velocity of the air passing over the heating surface. Where different sizes of the same design of heating surface are used, the curve is plotted on a square foot face area basis. Once this curve has been established, it requires only a very simple computation to determine what the outlet air temperature and Btu per hour will be for any given set of conditions. This method is only applicable to heating surfaces at constant temperatures, such as blast coils and the unit heaters on steam.

L. S. O'BANNON: I would like to point out that the success of this method depends largely upon the design of the nozzle. If the nozzle is too small, there will be too much resistance and the point representing the normal capacity of the unit may lie too far away from the points obtained by the tests and too much error may be introduced in extending the curve. If the nozzle is too large the velocity of the air may be too small to give an accurate final temperature reading. The design of the nozzle must avoid the two extremes and will depend upon the characteristics of the unit.

No. 857

THE MEASUREMENT OF THE FLOW OF AIR THROUGH REGISTERS AND GRILLES

By LYNN E. DAVIES[1], CHICAGO, ILL.

NON-MEMBER

The results of co-operative research between the Armour Institute of Technology, the A. S. H. V. E. Research Laboratory and the Ventilating Contractors Employers Association of Chicago.

FOR several years doubt has existed as to the proper or most accurate method of measuring the flow of air through grilles or registers. The fact that several methods were being used and that these various methods indicate widely different results gave rise to many controversies. Considerable financial loss resulted in many cases where the fulfillment of a contract was dependent upon these measurements. The laboratories of the Armour Institute of Technology, in cooperation with the Research Laboratory of the A. S. H. V. E. and the *Ventilating Contractors Employers Association of Chicago,* undertook an investigation of the problem in an attempt to arrive at a simple and accurate method of making such measurements.

It is apparent that a simple field test is desirable, one that is applicable to the ordinary ventilating system already installed and of such character as to cause little or no interference with the normal operation of the system. Hence, only two types of instruments need be considered at this time, the Pitot tube and the anemometer.

The nature of the air flow coming from a grille is such that no accuracy can be expected from the use of the Pitot tube. Furthermore, the velocities encountered are usually so low that it would necessitate the use of a gage several times more accurate than the usual type of portable instrument. Such sensitive gages are only practical for use in laboratories. It is therefore necessary to use the anemometer and the problem resolves itself into one of how to use this instrument and how properly to interpret its indications.

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Presented at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1930.

SUPPLY REGISTERS

Apparatus: The apparatus used in conducting these tests is shown in the accompanying sketches and photographs. It will be noted (Figs. 1 and 2) that in the original set-up the air from a centrifugal fan was passed through a transformation section into a 10-in. round duct 8 ft 3 in. in length. On the end of this round duct was a second transformation section 6 ft in length leading to a straight duct 4 ft in length and 2 ft square. In some of the tests the grilles were mounted on the end of this straight square duct. For the other

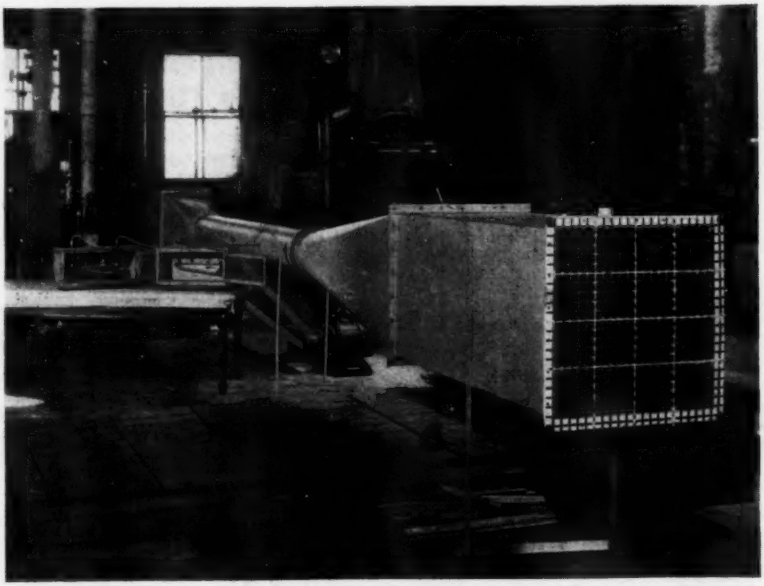

FIG. 1—VIEW SHOWING ORIGINAL SET-UP OF APPARATUS

runs an elbow or branched duct was connected to the end of the straight duct, and the grilles fastened on the opposite end of these units as shown in the photographs. (See Figs. 3 and 4.) The two branch ducts used were exactly alike in size and shape except that one contained a fixed vertical split starting two-thirds of the way from one side of the entrance to the duct, while the other contained a movable vertical split. Each of these branch ducts contained two outlets, 16 in. by 24 in. and 8 in. by 24 in. respectively. The entire system from fan to grille was made air-tight and particular care was taken to have the section of round duct smooth and uniform in size and shape.

After approximately half of the runs had been made, a large plenum chamber was added between the fan and the 10-in. duct. The entrance end of this chamber was fitted with grilles similar to those being tested. These served to

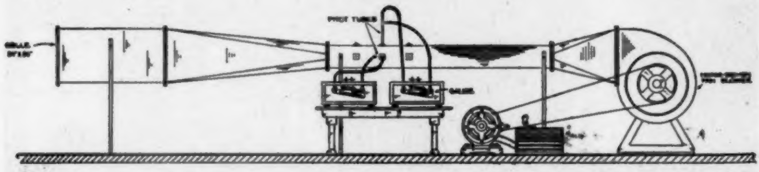

FIG. 2—ELEVATION SHOWING ORIGINAL APPARATUS AND FAN TRANSFORMATION, ROUND DUCT, SECOND TRANSFORMATION SECTION AND STRAIGHT DUCT

destroy the initial high velocity given to the air by the fan and to create a static condition in the discharge end of the chamber which would be favorable to the production of uniform parallel flow at the Pitot tubes. Although the Pitot tube readings were steadier after the addition of this chamber, the final results were not changed in any way.

Traverses were made in the round section 6 ft 3 in. from the blower end, with standard type Pitot tubes. Two traverses (horizontal and vertical) of 10 readings each were made for every anemometer traverse. The amount of air flow, as measured by these Pitot tubes, will hereafter be referred to as the *correct* or *actual* air flow. During the course of the work three different anemometers were used, each newly calibrated by the Bureau of Standards. All three instruments were of the same type.

Procedure: The surface of each grille was marked off into a number of

TABLE 1—DESCRIPTION OF GRILLES TESTED

1. Expanded metal lath having diamond shaped openings measuring 2 in. by 0.932 in. across the corners.	
Percentage of free area	72.0
2. Plain lattice work	
Size of openings	0.95 in. square
Spacing—center to center	1.19 in.
Thickness	1/16 in.
Percentage free area	63.25
3. Plain lattice work	
Size of openings	0.875 in. square
Spacing—center to center	1.125 in.
Thickness	1/16 in.
Percentage free area	60.5
4. Plain lattice work	
Size of openings	0.75 in. square
Spacing—center to center	1.00 in.
Thickness	1/16 in.
Percentage free area	56.25
5. Plain lattice work	
Size of openings	0.50 in. square
Spacing—center to center	0.75 in.
Thickness	1/16 in.
Percentage free area	44.44
6. Iron plate 3/8 in. thick round holes	
Diameter	0.73 in.
Percentage free area	38.2

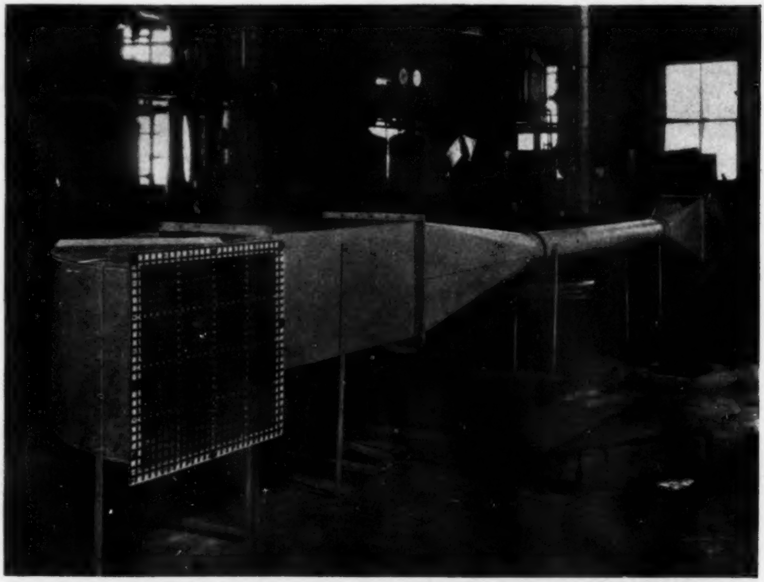

FIG. 3—VIEW SHOWING ELBOW AND GRILLE CONNECTED TO STRAIGHT DUCT

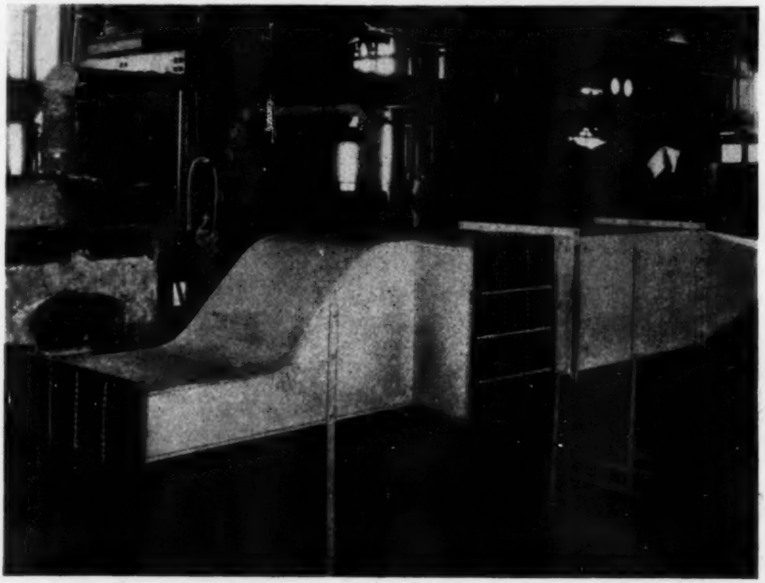

FIG. 4—VIEW SHOWING BRANCHED DUCT CONNECTED TO STRAIGHT DUCT

equal rectangular areas, 16 divisions in the case of the 24 $\times$ 24 in. grilles, 12 in the case of the 16 $\times$ 24 in. grilles, and 4 in the case of the smallest grilles used. During each run two readings were taken with the anemometer opposite the center of each small area. One of these readings in each case was taken with the anemometer held in contact with the surface of the grille, the other with the instrument held 3 in. away. In each case the average of the 16 readings obtained was corrected for instrument error and then multiplied by the gross area of the grille. Similarly each corrected average velocity was multiplied by the net free area. This resulted in five values for the air flow for each run, namely:

1. Anemometer reading in contact multiplied by gross area.
2. Anemometer reading in contact multiplied by net free area.
3. Anemometer reading 3 in. away multiplied by gross area.
4. Anemometer reading 3 in. away multiplied by net free area.
5. Quantity as measured by Pitot tubes.

By comparing the results obtained by the first four calculations, with the result of the Pitot tube traverse, it is possible to determine which method, if any, is the correct one for calculating the volume of air from the indicated velocities shown by the anemometer.

Discussion of Results: When the instrument is held in contact with the grille, the face of the anemometer is not subjected to a uniform flow of air over its entire area, but instead is exposed to a group of small jets divided by regions of dead, low velocity, or turbulent air. The portion of the area touched by the jets is probably approximately equal to the percentage of free area of the grille. To maintain that *Method 1* is correct is to claim, in effect, that an anemometer exposed to a variable velocity across its face indicates the true average velocity. This may appear to be a perfectly reasonable assumption, but those who are familiar with the action of anemometers know that they always indicate a value nearer to the maximum than to the minimum, and therefore somewhat higher than the average. The results obtained by this method, therefore, will probably be too high.

Method 2 would be correct if the anemometer indicated the maximum velocity striking the blades. This is not the case because in the regions of low velocities the vanes act as propeller, drawing the air forward, and this action naturally imposes a load on them which reduces their speed of rotation. This method may, therefore, be expected to give results which are too small.

Method 3 appears to be more nearly correct than the others if the instrument is held sufficiently far away to have the individual jets blend together into a solid stream of uniform velocity. The results indicate that 3 in. is not sufficient to accomplish this objective.

Method 4 does not require consideration, since it is apparent that a value lower than with *Method 2* is obtained and which is too small.

Table 1 gives a description of the types of grilles tested. Table 2 shows the results obtained when the anemometer was held in contact with the grille and the velocity indicated was multiplied by the gross area. It will be noted that the results obtained were from 19 per cent to 46 per cent too high, varying

somewhat with the velocity and to a much greater extent with the amount of free area. Table 3 shows the results obtained when the anemometer readings were multiplied by the net free area of the grille. In this case the results were from 13 per cent to 40 per cent too low. This is of particular interest because it is probable that this method has been used more than any of the others. Tables 4 and 5 show the results obtained from similar calculations based on readings taken 3 in. from the face of the grille. Here the errors ranged from + 10 per cent to + 36 per cent when the gross area was used, and from — 27½ per cent to — 44 per cent when the net area was used.

These results show plainly the urgent need for more accurate formulæ than any heretofore used. Referring again to Tables 2 and 3, a close observation

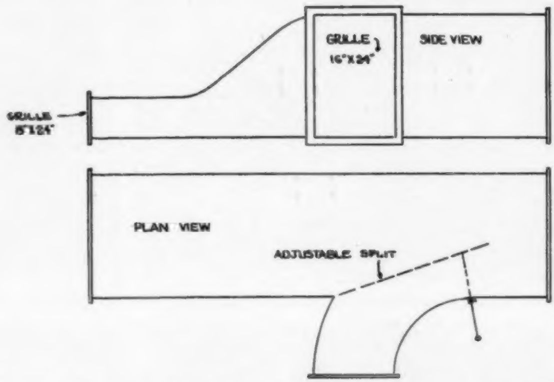

FIG. 5—PLAN AND SIDE VIEW OF ELBOW AND BRANCHED DUCT SHOWING ADJUSTABLE SPLIT

of the results shown in the right hand columns discloses the fact that for corresponding runs the percentage of error in the two cases is approximately equal. This suggested the use of a new formula:

$$cfm = \frac{V(A + a)}{2} \text{ or } \frac{VA(1 + p)}{2} \qquad (1)$$

in which

V = corrected average anemometer reading taken in contact with grille, in feet per minute.

A = gross area of grille, square feet.

a = net free area of grille, square feet.

p = percentage of free area of grille expressed as a decimal.

Table 6 shows the results of applying this formula. It will be noted that the maximum error was 5 per cent and that this could have been still further reduced had a coefficient of 0.98 been introduced into the formula. In view of the fact that the work, so far, has been confined to a limited number of types of grilles, it appears desirable that a coefficient C be introduced into the formula

TABLE 2—RESULTS OBTAINED WHEN ANEMOMETER WAS HELD IN CONTACT WITH THE GRILLE AND THE VELOCITY INDICATED WAS MULTIPLIED BY THE GROSS AREA

Position of Anemometer...In Contact With Grille
Area Used...Gross Area of Grille

Grille Used, Per Cent Free Area	Duct Used	Average Velocity Indicated by Anemometer, fpm	Air Flow Calculated from Anemometer cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow, Per Cent
72.0	*S*	242	968	800	121.0
63.25	*S*	757	3028	2467	123.0
		744	2976	2392	124.5
		540	2160	1691	127.5
56.25	*S*	766	3064	2330	131.5
		529	2116	1619	131.0
		268	1072	813	132.0
44.44	*S*	765	3060	2261	135.0
		538	2152	1545	139.0
		286	1144	805	142.0
63.25	*E*	713	2852	2390	119.0
		515	2060	1697	121.5
56.25	*E*	759	3036	2330	130.0
		530	2120	1615	131.0
44.44	*E*	756	3024	2200	138.5
		559	2236	1589	141.0
60.5	*BV*	318 167	849 222 } 1071	822.5	130.2
60.5	*BV*	218 352	581 469 } 1050	806.0	130.0
56.25	*B*	540 543	1440 724 } 2164	1685	129.0
56.25	*B*	440 410	1173 547 } 1730	1300	133.0
56.25	*B*	298 272	795 362 } 1157	860	134.5
44.44	*B*	364 350	971 467 } 1438	1026	140.0
44.44	*B*	245 242	653 323 } 976	669	146.0

to provide for a variation in results which may occur when different types of grilles are investigated. The formula will then become

$$cfm = \frac{CV(A + a)}{2} \text{ or } \frac{CVA(1 + p)}{2} \qquad (2)$$

It is the opinion of the author that the use of a constant value of 1.00 or 0.98 for *C* will introduce less error than that encountered from other sources such as errors in calculating free area, carelessness in making traverses, the difficulty of securing true average velocities when the distribution is poor, and

the use of anemometers which are inaccurate due to careless handling or long use without calibration. For securing the highest possible degree of accuracy, however, the results may be analyzed to determine what factors are most important in establishing the value of *C* to be used in this new formula. The average value of *C* obtained for each grille is as follows:

Grille	Per Cent Free Area	Average Value of *C*
1	72.0	0.956
2	63.25	0.995
3	60.50	0.958
4	56.25	0.976
5	44.45	0.985

The results appear to be somewhat erratic, but it is observed that No. 1 and No. 3 were tested only with relatively low air velocities, and while No. 2 was only tested with the higher velocities the results appear more logical. It is evident that the percentages of free area do not affect the value of *C* for the types of grilles investigated. A careful study of Table 6 will show that the highest values of *C* were secured with the highest velocities, and the lowest values with the lowest velocities. It appears then that the air velocity is a definite factor causing the variation in the value of the coefficient. The following values of *C* for the various velocities were obtained by means of the curve, Fig. 7:

Indicated Air Velocity fpm	*C*
150	0.952
200	0.957
300	0.967
400	0.977
500	0.985
600	0.992
700	0.998
800	1.000

In practice the velocity of the air coming from a register often varies greatly at different positions across its face. It is of interest to note, therefore, the amount of velocity variation encountered in this test. This is shown in Table 7. The last set of figures shown are of particular interest because of the exceptionally high variation. This was caused by the fact that the air had to pass from a 5½ in. $\times$ 24 in. opening at the split to a 16 in. $\times$ 24 in. discharge in a very short distance and while making a short radius turn. This resulted in the air issuing from the grille at a distinct angle and with a very low velocity along one side. In spite of these unfavorable conditions the coefficient agrees with the average value found on the branched duct with the fixed split, in which case a very favorable velocity distribution was obtained.

With some kinds of grilles it is difficult to estimate or measure accurately the percentage of free area. It is interesting to note the effect on the final result of an error of this kind. It is best illustrated by an example. Assume that a certain grille has a true free area amounting to 50 per cent of the gross area, but through some error is figured as 45 per cent free area. This represents an error in measurement amounting to 10 per cent and if used in the old formula (indicated velocity by free area) would result in a 10 per cent

Table 3—Results Obtained When the Anemometer Was Held in Contact With the Grille and the Readings Were Multiplied by the Net Free Area of the Grille

Position of Anemometer..In Contact with Grille
Area Used..Net Free Area of Grille

Grille Used, Per Cent Free Area	Duct Used	Average Velocity Indicated by Anemometer, fpm	Air Flow Calculated from Anemometer, cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow, Per Cent
72.0	*S*	242	696	800	87.0
63.25	*S*	757	1915	2467	78.0
		744	1884	2392	79.5
		540	1365	1691	80.5
56.25	*S*	766	1724	2330	74.0
		529	1190	1619	73.6
		268	604	813	74.3
44.44	*S*	765	1360	2261	60.0
		538	955	1545	61.8
		286	509	805	63.1
63.25	*E*	713	1806	2390	75.5
		515	1300	1697	76.5
56.25	*E*	759	1708	2330	73.5
		530	1192	1615	74.0
44.44	*E*	756	1345	2200	61.0
		559	992	1589	62.0
60.5	*BV*	318	514 } 649	822.5	78.9
		167	135		
60.5	*BV*	218	352 } 636	806.0	79.1
		352	284		
56.25	*B*	540	810 } 1216	1685	72.1
		543	406		
56.25	*B*	440	660 } 968	1300	74.4
		410	308		
56.25	*B*	298	447 } 651	860	75.8
		272	204		
44.44	*B*	364	430 } 636	1026	62.0
		350	206		
44.44	*B*	245	288 } 431	669	64.5
		242	143		

TABLE 4—RESULTS OBTAINED WHEN READINGS WERE TAKEN 3 IN. FROM THE FACE OF THE GRILLE AND THE VELOCITY MULTIPLIED BY THE GROSS AREA

Position of Anemometer....................3 In. from Grille
Area Used....................Gross Area of Grille

Grille Used, Per Cent Free Area	Duct Used	Average Velocity Indicated by Anemometer, fpm	Air Flow Calculated from Anemometer, cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow, Per Cent
63.25	*S*	680	2720	2392	114.0
		485	1940	1691	114.5
56.25	*S*	684	2736	2330	117.5
		474	1856	1619	115.0
44.44	*S*	712	2848	2256	126.0
		493	1972	1541	127.5
63.25	*E*	658	2632	2390	110.0
		479	1916	1697	113.0
56.25	*E*	703	2812	2330	121.0
		490	1960	1615	115.5
44.44	*E*	733	2932	2200	133.0
		540	2160	1589	136.0

error in volume of air in addition to the already large error due to the formula itself being incorrect. Using the new formula:

With 50 per cent area—

$$cfm = \frac{VA(1+p)}{2} = \frac{VA(1+0.50)}{2} = 0.75\ VA$$

With 45 per cent area—

$$cfm = \frac{VA(1+p)}{2} = \frac{VA(1+0.45)}{2} = 0.725\ VA$$

$$\text{Error} = 0.025\ VA$$

$$\text{Percentage Error} = \frac{0.025\ VA}{0.75} \times 100 = 3.33 \text{ per cent}$$

Thus it is apparent that the 10 per cent error in measurement only causes a 3⅓ per cent error in the value of *cfm*.

EXHAUST REGISTERS

Apparatus: For the purpose of investigating the performance of the anemometer on exhaust registers the same apparatus was used except that the test duct was connected to the suction side of the fan. All of the work was done with the straight duct as it appeared that the shape of the duct would have little if any effect in this case.

Method: The general procedure was the same as before except that all of the anemometer traverses were made in contact with the grille. The nature of the air stream surrounding an exhaust grille is very different from that surrounding a supply grille. The air around the exhaust register is being

drawn in from all sides as well as straight from the front, so that an anemometer traverse made in front of the opening but some distance away would only show a portion of the total flow. The percentage of the total thus effecting the anemometer would vary greatly with the total area of the grille, its shape and the nature of the surrounding wall surface. It therefore appears essential that the traverse be made as close to the surface of the grille as practical.

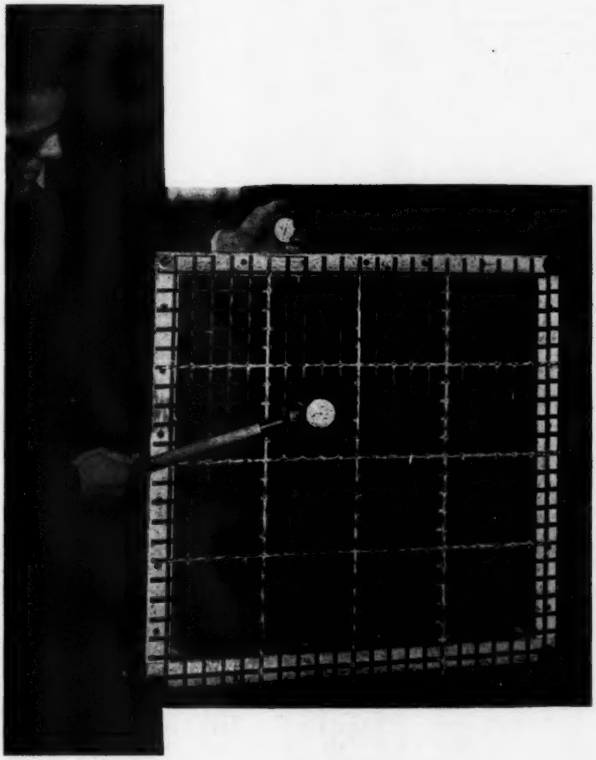

Fig. 6—Method of Taking Traverse

Discussion of Results

Table 8 shows the results obtained when the anemometer readings were multiplied by the net free area of the grille. It will be noted that the error amounted to from 7.3 per cent to 51.6 per cent, a much wider variation than that obtained with supply grilles when the same formula was used. Table 9 shows the results of applying the new formula (1). It can be seen that the results vary more than was the case with the supply systems. The results

TABLE 5—RESULTS OBTAINED WHEN THE READINGS WERE TAKEN 3 IN. FROM THE FACE OF THE GRILLE AND THE VELOCITY INDICATED MULTIPLIED BY THE NET FREE AREA OF THE GRILLE

Position of Anemometer....................3 In. from Grille
Area Used....................Net Free Area of Grille

Grille Used, Per Cent Free Area	Duct Used	Average Velocity Indicated by Anemometer, fpm	Air Flow Calculated from Anemometer, cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow, Per Cent
63.25	*S*	680	1720	2392	72.0
		485	1228	1691	72.5
56.25	*S*	684	1539	2330	66.0
		474	1066	1619	66.0
44.44	*S*	712	1265	2261	56.0
		493	875	1545	56.5
63.25	*E*	658	1665	2390	69.5
		479	1210	1697	71.0
56.25	*E*	703	1582	2330	68.0
		490	1102	1615	68.5
44.44	*E*	733	1301	2200	59.2
		540	960	1589	60.5

TABLE 6—RESULTS OF APPLICATION OF FORMULA (1)

Position of Anemometer....................In Contact with Grille
Area Used....................$\frac{\text{Gross Area} + \text{Net Area}}{2}$

Grille Used, Per Cent Free Area	Duct Used	Average Velocity Indicated by Anemometer, fpm	Air Flow Calculated from Anemometer, cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow, Per Cent	*C*
72.0	*S*	242	835	800	104.4	0.957
63.25	*S*	757	2471	2467	100.1	0.999
		744	2430	2392	101.6	0.985
		540	1762	1691	104.1	0.961
56.25	*S*	766	2394	2330	102.5	0.976
		529	1653	1619	102.1	0.978
		268	838	813	103.0	0.971
44.44	*S*	765	2210	2261	97.7	1.021
		538	1553	1545	100.6	0.994
		286	825	805	102.5	0.977
63.25	*E*	713	2329	2390	97.5	1.027
		515	1680	1697	99.1	1.009
56.25	*E*	759	2372	2330	101.8	0.983
		530	1656	1615	102.5	0.977
44.44	*E*	756	2184	2200	99.5	1.005
		559	1614	1589	101.8	0.983
60.5	*BV*	318 167	$\left.\begin{matrix}682\\178\end{matrix}\right\}860$	822.5	104.2	0.959
60.5	*BV*	218 352	$\left.\begin{matrix}466\\377\end{matrix}\right\}843$	806.0	104.2	0.959

TABLE 6—RESULTS OF APPLICATION OF FORMULA (1)—*Continued*

Position of Anemometer...In Contact with Grille
Area Used..$\frac{\text{Gross Area} + \text{Net Area}}{2}$

Grille Used, Per Cent Free Area	Duct Used	Average Velocity Indicated by Anemometer fpm	Air Flow Calculated from Anemometer cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow, Per Cent	*C*
56.25	*B*	540 543	1152 565 } 1690	1685	100.3	0.997
56.25	*B*	440 410	916 427 } 1343	1300	103.0	0.971
56.25	*B*	298 272	621 283 } 904	860	105.0	0.953
44.44	*B*	364 350	700 336 } 1036	1026	101.0	0.990
44.44	*B*	245 242	470 233 } 703	669	105.0	0.953

can be more clearly visualized by Fig. 8 on which the value of *C* has been plotted against the velocities shown by the anemometer for each grille.

The great difference between the *38.2 per cent* and the *44.44 per cent* curves and the *63.25 per cent* and *72 per cent* curves indicate that, unlike the case of

TABLE 7—VARIATION IN VELOCITY AT DIFFERENT POSITIONS ACROSS THE FACE OF THE GRILLE

Duct Used	Grille Used, Per Cent Free Area	Variation of Velocity Across Grille Expressed as Ratio Between Maximum and Minimum Velocities for Any One Run. Max. Variation	
Straight	72.0 63.25 56.25 44.44	2.04 1.53 1.38 1.24	
Elbow	63.25 56.25 44.44	1.62 1.35 1.27	
		Small Branch	Large Branch
Branched } Fixed Split }	56.25 44.44	1.06 1.11	1.25 1.085
Branched[a] Adjustable Split } No. 1....................	60.25	1.14	1.32
Branched[a] Adjustable Split } No. 2....................	60.25	1.10	5.94

[a]For Run No. 1 the split was moved over to a point 2 in. from the wall leading to the small outlet.
For Run No. 2 the split was moved to a position 5½ in. from the wall leading to the large outlet.

supply grilles, the values of C vary not only with the velocity, but also with the percentage of free area and the type of grille. The 38.2 per cent grille had solid metal between the openings measuring approximatly 0.45 in. $\times$ 1.2 in. which is more than is encountered in the average grille, while the 72 per cent grille was of expanded metal lath measuring approximately ⅛ in. across the frets and having the edges pointed in the direction of the air flow. It is probable that these two styles represent the greatest extremes likely to be encountered in practice.

For the velocity range from 150 *fpm* to 700 *fpm* the maximum value of C is 1.145 while the minimum value is 0.90. If a constant value of 1.02 is used, there will be a possible maximum error of 12.5 per cent although usually the error will be considerably less.

Since the value of C does not seem to follow any rational law as to the type of grille, it appears that the only alternative is to draw an average curve as shown by the dotted line of Fig. 8. This has been drawn somewhat higher than half way between the extreme curve, but appears to be a better one for the average type of grille. From this curve, the factors shown in the following table can be obtained:

Indicated Air Velocity fpm	C
150	0.993
200	1.005
300	1.028
400	1.049
500	1.067
600	1.078
700	1.084

When these values are used there will be a possible error of approximately 10 per cent but will result only under extreme circumstances. In the majority of cases the error will be less than 3 per cent.

Table 10 shows the results obtained when the indicated velocities were multiplied by the gross area of the grille. These show that while there is a variable error, the amount of variation is much less than it was with supply grilles, particularly if values for equal velocities are compared. The final column shows the value of the coefficient K which may be used, giving the formula:

$$cfm = KVA \qquad (3)$$

in which

V = Indicated velocity in feet per minute.

A = Gross area, square feet.

Fig. 9 shows the values of K plotted against velocity for various grilles. It will be noted that if a constant value of 0.8 is used for K the maximum error will be 7½ per cent while if an average curve is drawn, as before, and a factor

TABLE 8—RESULTS OBTAINED WHEN ANEMOMETER READINGS WERE MULTIPLIED BY THE NET FREE AREA OF THE GRILLE

Position of Anemometer..In Contact with Grille
Area Used..Net Free Area

Grille Used, Per Cent Free Area	Average Velocity Indicated by Anemometer, fpm	Air Flow Calculated from Anemometer Reading, cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow Per Cent
72.0	160	461	497	92.7
	448	1290	1478	87.3
	583	1676	1942	86.2
63.25	136	344	425	80.9
	180	455	575	79.1
	195	494	627	78.6
	261.6	664	845	78.5
	278.7	705	905	78.0
	332	840	1108	76.0
	354.5	895	1152	77.6
	399	1010	1360	74.2
	413	1047	1398	74.9
	418	1059	1408	75.1
	490	1240	1661	74.6
	494	1250	1652	75.6
	556	1410	1910	73.9
	559.5	1412	1903	74.2
60.5	286	693	945	73.4
	435.5	1053	1460	72.1
56.25	155	348	487	71.5
	257	577.5	830	69.5
	338	761	1126	66.0
	411	925	1390	66.5
	454	1020	1522	67.0
	565	1270	1925	65.9
44.44	145	257	428	60.0
	200	355.5	610	58.3
	214	380	639	59.5
	226	401	684	58.8
	239.5	425	720	59.0
	265	470	806	58.2
	291	516	880	58.8
	338	600	1038	58.0
	374	665	1155	57.5
	392.5	696.5	1220	57.1
	449	798	1410	56.6
	485	861	1530	56.4
	501	890	1575	56.5
	586	1040	1882	55.5
	595	1057	1898	55.7
	605	1072	1910	56.3
	607	1078	1935	55.6
	618.5	1098	1940	56.5
38.2	260	397	800	49.6
	500	904	1870	48.4

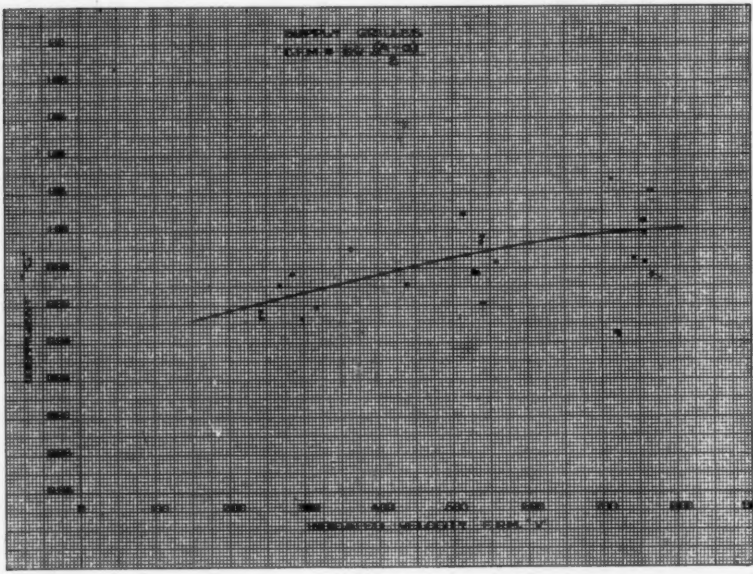

FIG. 7—CURVES SHOWING RELATION BETWEEN INDICATED VELOCITY AND COEFFICIENT C FOR SUPPLY GRILLES

varying with the velocity applied, the resulting error will be reduced to a maximum of less than 4 per cent. The values of K in this case would be as follows:

Indicated Air Velocity fpm	K
150	0.762
200	0.772
300	0.789
400	0.806
500	0.820
600	0.828
700	0.832

It is possible that a further study of these figures may disclose another formula involving a constant coefficient with smaller maximum errors than those encountered with the two considered. For example, the following might prove satisfactory.

$$cfm = C'V \ (0.8A + 0.2a)$$

A formula of this character would, of course, not eliminate the small change in C caused by the change in velocity. Any formula that would attempt to

TABLE 9—APPLICATION OF RESULTS TO FORMULA (2)

Position of Anemometer...In Contact with Grille

Area Used...$\frac{\text{Gross Area + Net Area}}{2}$

Grille Used, Per Cent Free Area	Average Velocity Indicated by Anemometer, fpm	Air Flow Calculated from Anemometer Reading, cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow, Per Cent	*C*
72.0	160	550.5	497	111.0	0.901
	448	1541	1478	104.1	0.959
	583	2004	1942	103.1	0.97
63.25	136	444	425	104.1	0.96
	180	587.5	575	102.1	0.978
	195	637	627	101.7	0.983
	261.6	855	845	101.1	0.989
	278.7	909.9	905	100.6	0.994
	332	1084	1108	98.1	1.019
	354.5	1156.5	1152	100.4	0.996
	399	1303	1360	96.0	1.040
	413	1349.5	1398	.....	1.027
	418	1365.5	1408	97.0	1.030
	490	1600	1661	96.3	1.038
	494	1613	1652	97.5	1.025
	556	1817	1910	95.2	1.050
	559.5	1826	1903	95.9	1.041
60.5	286	918.5	945	97.0	103.1
	435.5	1397.5	1460	95.6	104.3
56.25	155	484	487	99.4	1.006
	257	802.7	830	96.7	1.031
	338	1056.5	1126	93.6	1.068
	411	1284.5	1390	92.4	1.081
	454	1418.0	1522	93.0	1.076
	565	1765	1925	91.6	1.09
44.44	145	418.5	428	97.8	1.021
	200	578	610	94.7	1.055
	214	618	639	97.0	1.031
	226	652.5	684	95.5	1.048
	239.5	692	720	96.1	1.040
	265	765	806	94.9	1.052
	291	840	880	95.5	1.047
	338	976	1038	94.2	1.061
	374	1080.5	1155	93.8	1.066
	392.5	1133.5	1220	93.1	1.072
	449	1297	1410	92.0	1.086
	485	1400.5	1530	91.5	1.091
	501	1447	1575	91.8	1.089
	586	1692	1882	90.0	1.111
	595	1718.5	1898	90.5	1.105
	605	1746	1910	91.5	1.091
	607	1753	1935	90.7	1.103
	618.5	1786	1940	92.1	1.085
38.2	260	718.5	800	89.9	1.110
	590	1632	1870	87.5	1.141

correct for this variable would be a quadratic containing a factor of the form $(V+V^n)$ in which n would be a value less than 1. It is the author's opinion that the use of any such complicated formula is unwarranted under the circumstances.

The results of these tests are summarized in Table 11. All of the curves

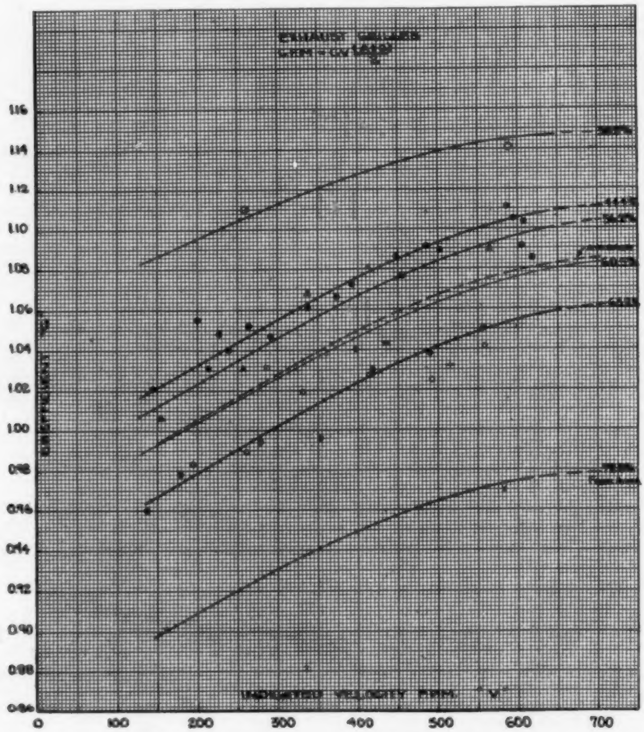

FIG. 8—CURVES SHOWING RELATION BETWEEN INDICATED VELOCITY AND COEFFICIENT C FOR VARIOUS PERCENTAGES OF FREE AREA FOR EXHAUST GRILLES

for the various coefficients appear to approach the horizontal as the velocity is increased. Lack of proper equipment has prevented verifying this fact, but it is expected that this can be done in the near future.

It is frequently asked if it is necessary to apply an anemometer correction to each individual reading when a number of spot readings are taken as was the case in these experiments. If the actual velocity is plotted against the indicated velocity, the curve obtained will be a straight line in most cases, and it is therefore mathematically correct to use the average of the various velocities

TABLE 10—RESULTS OBTAINED WHEN INDICATED VELOCITIES WERE MULTIPLIED BY THE GROSS AREA OF THE GRILLE

Position of Anemometer..In Contact with Grille
Area Used...Gross Area

Grille Used, Per Cent Free Area	Average Velocity Indicated by Anemometer fpm	Air Flow Calculated from Anemometer Reading, cfm	Actual Air Flow, cfm (Pitot Tube)	Ratio of Calculated Flow to Actual Flow, Per Cent	K
72.0	160	640	497	1.288	0.776
	448	1792	1478	1.21	0.825
	583	2332	1942	1.20	0.835
63.25	136	544	425	1.277	0.783
	180	720	575	1.250	0.800
	195	780	627	1.241	0.804
	261.6	1046.5	845	1.238	0.808
	278.7	1115	905	1.230	0.811
	332	1328	1108	1.200	0.833
	354.5	1418	1152	1.229	0.815
	399	1596	1360	1.172	0.851
	413	1652	1398	1.181	0.845
	418	1672	1408	1.19	0.840
	490	1960	1661	1.18	0.849
	494	1976	1652	1.192	0.837
	556	2224	1910	1.167	0.857
	559.5	2238	1903	1.172	0.852
60.5	286	1144	945	121.1	0.825
	435.5	1742	1460	119.2	0.837
56.25	155	620	487	127.0	0.786
	257	1028	830	123.5	0.809
	338	1352	1126	118.1	0.845
	411	1644	1390	118.1	0.845
	454	1816	1522	119.0	0.839
	565	2260	1925	117.2	0.852
44.44	145	580	428	135.3	0.738
	200	800	610	131.0	0.762
	214	856	639	134.1	0.745
	226	904	684	132.1	0.756
	239.5	958	720	133.0	0.751
	265	1060	806	131.2	0.761
	291	1164	880	132.2	0.755
	338	1352	1038	130.5	0.766
	374	1496	1155	129.5	0.771
	392.5	1570	1220	129.0	0.775
	449	1796	1410	127.1	0.785
	485	1940	1530	127.0	0.787
	501	2004	1575	127.1	0.785
	586	2344	1882	124.8	0.801
	595	2380	1898	125.2	0.797
	605	2420	1910	126.9	0.789
	607	2428	1935	125.5	0.796
	618.5	2474	1940	127.3	0.785
38.2	260	1040	800	130.0	0.769
	590	2360	1870	126.0	0.794

as read and to then apply a single correction factor. Furthermore, it is possible to extend this straight line below the lowest point obtained by actual calibration with entirely consistent results.

The question most frequently asked is that regarding the feasibility of making a traverse by moving the instrument over the face of the grille, thus securing a single reading instead of a number of spot readings. Check readings

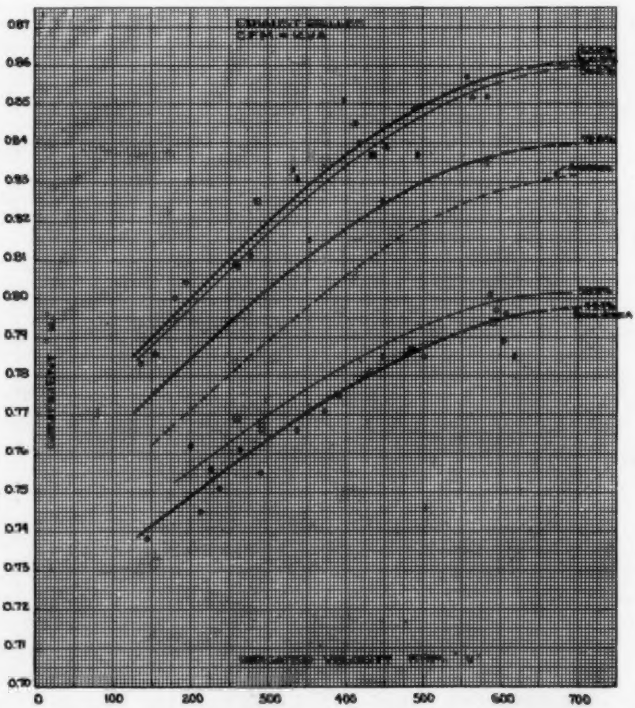

FIG. 9—CURVES SHOWING RELATIONSHIP BETWEEN COEFFICIENT *K* AND INDICATED VELOCITY FOR VARIOUS PERCENTAGES OF FREE AREA FOR EXHAUST GRILLES

were frequently made in this manner throughout these tests and quite satisfactory results were secured. The greatest error is likely to result when there is a wide variation in velocity across the face of the grille. When the instrument is observed to be slowing up in a region of low velocity there is a natural tendency to move it quickly to a region of higher velocity. For this reason the moving traverse is liable to give results that are somewhat too high

Table 11—Summary of Results

Average Indicated Velocity fpm	$\text{cfm} = \frac{CV(A+a)}{2}$ Supply Grilles C	$\text{cfm} = \frac{CV(A+a)}{2}$ Exhaust Grilles C	cfm = KVA Exhaust Grilles Only K
150	0.952	0.993	0.762
200	0.957	1.005	0.772
300	0.967	1.028	0.789
400	0.977	1.049	0.806
500	0.985	1.067	0.820
600	0.992	1.078	0.828
700	0.998	1.084	0.832
800	1.000		
Factor for average use	low vel. 0.97 high vel. 1.00	1.02	0.8

Note:
A = gross area of grille—square feet
a = net free area of grille—square feet
Each table of coefficients approaches a constant value as the velocity is increased so that it is believed that the highest value shown in each case can be used for all higher velocities with very slight error.

under these circumstances. This can be at least partly overcome if, before starting to run, a mental note is made of the time which should have elapsed when certain points in the traverse are reached.

With grilles the size of those used in these tests, such a run should be of at least two minutes duration unless the velocity is exceptionally uniform throughout the entire region traversed.

It is the opinion of the author that the data so far secured will make it possible to make anemometer measurements with a far greater degree of accuracy than has been possible in the past.

DISCUSSION

S. H. Givelber: What practical method would be recommended by the author to determine the net free area of an ornamental iron grille which is ordinarily found on various types of jobs?

Perry West: I would like to ask Professor Davies what method he used to adjust the anemometer to a distance of 3 in. from the grille and to maintain that adjustment.

John Howatt: At a meeting of the Illinois chapter a contractor asked me how we tested air deliveries in our school work. I told him that for 15 years we had been taking the readings 2 in. from the face of the register and considering the gross area of the register. It seems now that we were not getting correct results. I know in the future we will change our methods.

It is interesting to see that the larger error takes place in the lower velocities. Professor Davies indicates that with an elbow at the end of the riser, as there nearly always is on school work, there is a very low velocity at the short end of the elbow, even if we use splitters at the outlet. We have found (and I

think every one else has) that the anemometer shows a very high velocity at the outside of the curve and a very low velocity at the inside, and any factor given as a constant to correct the readings will not accurately make the correction unless the velocity is nearly uniform all over the register.

Whatever has been decided as the proper way of measuring air velocities by an anemometer on the register face should be adopted as a standard. A controversy was started because there was no accepted standard for measuring velocities at registers. The architects, the owners and the engineers must all come to an understanding as to what is the best and proper way of measuring these air velocities so that every one will know what to expect.

MARGARET INGELS: Did the construction of the anemometer introduce an error when the instrument was turned with the dial towards the exhaust to take care of that backward flow of the air? The anemometer would not be exactly flush; but would probably be a quarter or a half inch from the face of the register.

J. R. MCCOLL: I would like to ask Professor Davies how much time he gave to the position of the anemometer in one part of the steam or grille, or did he keep it moving all over the grille?

L. E. DAVIES: The first speaker touched a sore spot when he asked how to measure the area of the cast-iron grilles. I expected that question; and I will admit that I have not any convenient, simple method to recommend at the present time. Of course, if you want to go to great pains to remove the grille, there are different ways that may be done, but a simple method, without disturbing the setting, is not the easiest thing in the world to find. Possibly some months from now we may be able to answer that question.

I fitted to my anemometer a very small piece or strip of brass, small enough so that it wouldn't in any way interfere, but large enough so that I would have something to rest against so as to hold my space uniform. That, you see, was in the direction of the flow so that it should not in any way interfere with the nature of the flow. I had it arranged so that I could quickly remove it when I wanted to make my contract readings.

That, incidentally, is an objection to the method of taking a traverse some distance off. The average man doing it would not have any such space arranged and where you might specify 2 in. he might actually hold it anywhere from ½ to 6 in.

One man mentioned the fact that our factor should not be expected to be accurate when there was a wide variation in velocity such as at the elbow. What I wanted to bring out was the fact that apparently we were able to have quite a wide variation in that velocity and furthermore to have the velocity air coming out at quite an angle without seriously affecting the results. I do not know if I emphasized that, but in the case of the branch duct, where we had this variable split moved over close to the side where the large opening was located, the air was coming from the grille at quite a distinct angle and of the twelve divisions two of them had very, very low velocities. I believe that this indicates the fact that the coefficient under such circumstances does not differ from the normal value. You could not detect it on the curve. It indicates you need not worry very much except, of course, in extreme cases where you might have air flowing back at a high velocity; but in cases where the air is

coming out at all, or is only going back at a very low velocity, I would say take your traverse in the normal manner and you will be very close to right.

Now, regarding the effect on the exhaust measurements of the anemometer being out a little way, since due to its construction you are not able to get it up in contact. That is one of the first things that occurred to me when I started to do the work on exhaust grilles and found the results were differing a little bit from what I had hoped for.

On supply, I found that it would have made very little difference, but in that case, of course, we were not worried by it because the glass was outside and did not interfere with our getting in contact. It would be better if it was the other way round. I found that the readings taken on the supply an inch or so away differed very, very little from the readings taken in contact, but I believe this would not be the same on the exhaust. So in my instrument I removed the glass for a few runs to see whether I could get any different result; so that I could let it get up just as close as possible. There was quite a bow on the glass of instrument so it made some difference in its position. I found I could not detect the difference in the results.

Regarding the nature of taking the readings, when I started my work I used the flying start method. I had been in the habit of using that. I simply let the instrument go continuously and started and stopped the stop watch. It makes the figuring a little more difficult, but it is just a question of being used to it; that is all. I took about half minute readings, nothing less than half a minute. But, of course, the time was not uniform because I was going by a uniform number of revolutions of the dial rather than a uniform period of time. When it came to the exhaust measurements, I had to go to the other method of taking half minute readings and starting and stopping the anemometer.

Many times they do not care to go to the trouble of taking 16 or more spot readings. They want to take a moving traverse. In a large number of cases I have done that and I have had the students do it, thinking that perhaps being more experienced in it I might be able to get satisfactory results whereas another man with less experience would not. So during the last two or three months I have been having students make runs. I do not mean any of these data are students' work, but they have been doing some work on it also and been able to check my work very nicely. In each case I had them make a moving traverse, as well as the spot traverse, and the results were seldom more than three per cent different. They would sometimes be a little low and sometimes a little high.

I would say that the greatest error in such a case would come when there was a wide variation in velocity. For one thing, there is a tendency to want to hustle past the points of low velocity. Of course, if you can overcome that tendency, why it is going to help matters a good deal. It is a natural feeling. I have had it myself many times. "Let us not waste any time here. It is not moving much anyway and just go on." So that I would say that you are more likely to get a reading too high than too low but if you can overcome that tendency, you can check quite closely.

Another thing, though, I think that when they take moving traverses many times they do it too fast. They try to do it in half a minute over a large surface. I would say that you should not devote less than 2 min on a grille

the size of the ones that I used, that is two feet square. That is what I have done.

I think I have touched on the different points.

MR. MARSHALL: So far during this discussion we have taken it for granted that the anemometer is in calibration. Those of us who have used anemometers know that they are very easily put out of calibration. I think that it is very important to emphasize the fact that an anemometer should be calibrated before a test is made. Very often the instrument is in the hands of people who are not familiar with the delicacy of the instrument. I know I have gone out on jobs with anemometers and the contractor also had an anemometer on the job and it did not coincide with the corrections that I had for my instrument. In other words, there was sometimes a variation as high as 20 per cent in the accuracy of the instruments used.

No. 858

RATING OF HEATING BOILERS BY THEIR PHYSICAL CHARACTERISTICS

By C. E. Bronson,[1] Kewanee, Ill.

MEMBER

THIS paper is presented in explanation of the Code for the Rating of Low Pressure Heating Boilers adopted by the members of the *Steel Heating Boiler Institute*. The complete text of the Code has been submitted to the Society's Committee on Rating Low Pressure Steam Heating Boilers and will be found at the end of this paper.

This method of rating boilers is offered as an alternate to the method of determining ratings by means of test characteristics. The proposed method has the advantage of representing the size of the boiler by a single number, which has a certain and definite relation not only to the load for which the boiler is selected, but also to the operating characteristics of the boiler itself. This single number rating is easily determined from measurable proportions of the boiler and obviates the necessity for a long and expensive program of tests.

There has been much discussion on the question of boiler ratings and the relation of rating to selection. Attempts have been made to separate consideration of the two but the author believes they should be inseparable. The rating should be a single number and should have a definite relation to the load imposed on the boiler. This method has been successfully used by the manufacturers of steel heating boilers for many years. The rating of a boiler by any other method than the one proposed would cause confusion to those accustomed by long years of practice to selecting boilers from a single number rating.

If a single number rating is assigned to a boiler it must be an indication of capacity. The capacity of a boiler is dependent upon two main variables.

1. Rate of fuel burning.
2. Ability to absorb heat.

The rate at which fuel may be burned is dependent upon the area of the grate, the kind of fuel and the available draft. If the draft required or the chimney size is given for the ordinary grades of fuel used, it would be possible to assign a value for the rated capacity of a boiler based on the grate area. This considers only hand-fired solid fuel burning boilers.

The ability to absorb heat is dependent upon the amount of heating surface interposed between the grate and the chimney. Therefore, it is logical to assign

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Presented at the 36th Annual Meeting of the American Society of Heating and Ventilating Engineers, Philadelphia, Pa., January, 1930.

a uniform value of capacity based on total heating surface. Then it is possible to assign a capacity value or rating to a heating boiler based on:

1. Heating surface.
2. Grate area.

The proposed Code defines the rating as the heating surface multiplied by a factor of 14. There is a precedent in the choice of such a factor. Ratings for power boilers have been made for years on the basis of 10 sq ft of heating surface per boiler horse-power. A boiler horse-power is defined as the evaporation of 34.5 lb of water per hour, from and at 212 F. This is equal to the delivery of 34.5 $\times$ 971.7 (latent heat at 212 F) or 33,524 Btu per hour. Expressed in square feet of radiation, a boiler horse-power is equal to $\frac{33,524}{240}$ or 139.7, practically 140. With 10 sq ft of heating surface per boiler horse-power of rating, the ratio of rating to heating surface becomes 14, the figure selected.

The practice in former years in selecting boilers based on boiler horse-power was to allow one boiler horse-power for every 100 sq ft of radiation leaving a margin of 40 per cent to take care of losses in mains, risers and returns. This is equivalent to making the boiler rating 10 times the heating surface. Since 40 per cent is rarely required to care for losses due to mains, etc., present boiler ratings have been assigned on ratios varying from 10 to 12 times the heating surface, the manufacturer recommending that losses due to mains, risers and returns need not be figured but that the boiler be selected on the basis of installed radiation. Other manufacturers have assigned still higher ratings based on values as high as 14 times the heating surface and recommending that the entire normal load be used in selecting the proper size of boiler. A uniform method is desirable, consequently the proposed Code recommends in placing the rating at 14 times the heating surface, that the total normal load be used in selection. By this method the boiler rating is equal to the boiler output required for the normal heat demand.

Fig. 1 shows how present boiler ratings are based on heating surface. Although more than 800 boiler sizes are plotted, there are not this number of points indicated as some of the values are coincident. The curve shown gives values of ratings equal to 14 times the heating surface. Similarly, Fig. 2 indicates the relation between grate area and rating for the same boilers. The curves shown are plotted from the formulæ given in the Code.

In order to give a clear conception of the Code requirements, Table 1 shows the minimum heating surface and grate areas for boilers of various ratings. For each rating, values of combustion rate, fuel charge and firing period are given for the condition when the boiler output is equal to the rating.

The formulæ for minimum grate area were developed after due consideration had been given to the various factors involved. In general, small boilers installed in residences are connected to chimneys of 30 or 40 ft in height. The draft available from such chimneys is comparatively small in value. Long periods of time without attention to the fire are desirable. In hand-fired boilers the grate must be utilized partly as fuel storage space. Consequently, rates of combustion for small boilers must be kept low in value. Larger boilers have higher chimneys, are fired more frequently and therefore may be allowed higher combustion rates.

While more grate area should be allowed for small boilers than for large boilers, it seemed that the rating of boilers of acceptable proportions increased approximately according to the square of the grate area. This can be readily noted by reference to Fig 2. The formulæ cover two ranges of rating values

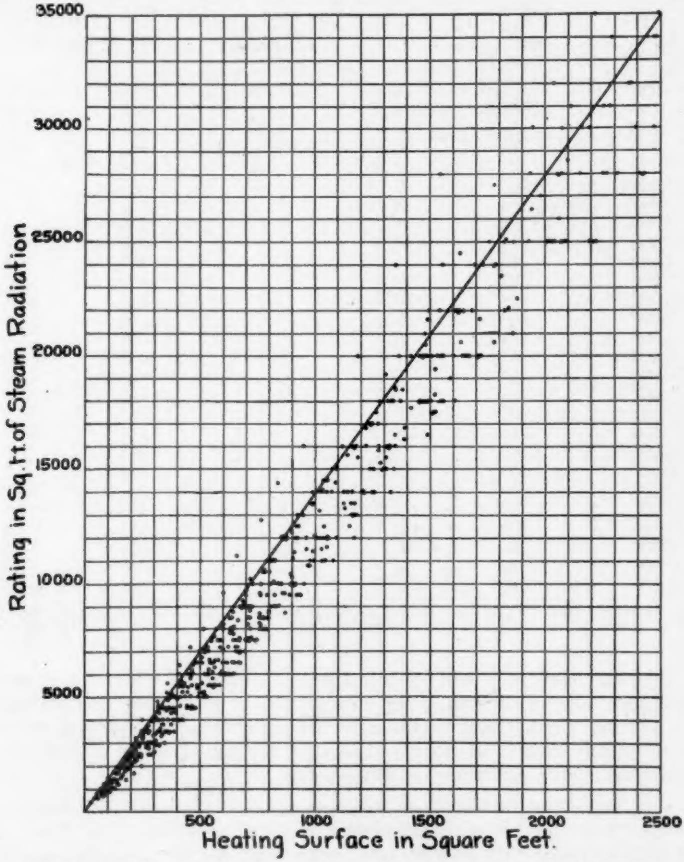

FIG. 1. RELATION OF PRESENT BOILER RATINGS TO HEATING SURFACE

as it was considered easier to use even instead of fractional exponents. Resulting grate areas calculated by the formulæ were also checked with the allowances made by the other authorities[3] and found to agree quite closely.

There is no priming limitation mentioned in this Code. It has always been tacitly understood that a boiler built for producing steam means one capable

[3] *Net loads for Heating Boilers issued* by the Heating and Piping Contractors National Association in April, 1929.

TABLE 1. MINIMUM HEATING SURFACE AND GRATE AREA REQUIREMENTS WITH CORRESPONDING FIRING CHARACTERISTICS WHEN BOILER OUTPUT IS EQUAL TO RATING

Boiler Rating — Sq Ft Steam Radiation	Heating Surface — Sq Ft	Grate Area — Sq Ft	Combustion Rate[a] — Lb Coal per Sq Ft Grate Area per Hour	Fuel Charge — Lb	Firing Period — Hours
300	21.4	1.98	4.66	78	8.5
400	28.6	2.80	4.39	111	9.0
500	35.7	3.42	4.49	135	8.8
750	53.6	4.64	4.97	184	8.0
1,000	71.4	5.60	5.49	222	7.2
1,500	107	7.14	6.47	283	6.1
2,000	143	8.39	7.34	332	5.4
2,500	178	9.5	8.1	376	4.9
3,000	214	10.5	8.8	415	4.5
4,000	286	12.2	10.1	483	3.9
5,000	357	14.4	10.7	570	3.7
6,000	429	16.4	11.2	650	3.5
7,000	500	18.1	11.9	716	3.3
8,000	571	19.7	12.5	780	3.2
9,000	642	21.1	13.1	835	3.0
10,000	714	22.5	13.7	890	2.9
12,000	857	25.0	14.8	990	2.7
14,000	1,000	27.3	15.8	1,080	2.5
16,000	1,143	29.4	16.7	1,164	2.4
18,000	1,285	31.4	17.6	1,243	2.2
20,000	1,430	33.2	18.5	1,314	2.1
22,500	1,608	35.3	19.6	1,397	2.0
25,000	1,780	37.4	20.6	1,480	1.9
27,500	1,963	39.3	21.5	1,555	1.8
30,000	2,140	41.1	22.4	1,626	1.8
32,500	2,320	42.9	23.3	1,700	1.7
35,000	2,500	44.5	24.2	1,760	1.6

a Combustion rate calculated for 65 per cent efficiency and 12,000 Btu coal. Fuel charge based on 10 in. depth of coal weighing 47.5 lb per cu ft.

of furnishing steam only and not a mixture of steam and water. A limitation of moisture content of 2 or 3 per cent would be acceptable and reasonable. Since a boiler which would furnish more than a reasonable amount of moisture with the steam could be no longer classed as a steam boiler, it is believed unnecessary to comment further on this point.

Rating, according to this Code, is made equal to the estimated design load or normal load and is the recommended basis of selection. The three items considered as making up the estimated design load are the same as items *A, B* and *C* outlined in Section V of the A.S.H.V.E. Code of Minimum Requirements for the Heating and Ventilation of Buildings. No reference has been made to starting load or the load imposed on the boiler due to warming up cold mains and radiation. The starting load is considered an overload on the boiler above its rating and is taken care of by the boiler manufacturer in the specification of chimney for each boiler. Under present methods of selecting steel boilers and with chimneys as recommended, the maximum output exceeds the total requirements as specified in the Code of Minimum Requirements. In other words, there is greater margin allowed between maximum capacity

developed and normal load. Maximum capacity is dependent primarily on available draft and efficiency of the heating surface.

The constants used in the Code for the determination of rating are admittedly arbitrary. Yet these constants have been selected with due regard to controlling factors. Furthermore, the boiler manufacturers themselves have

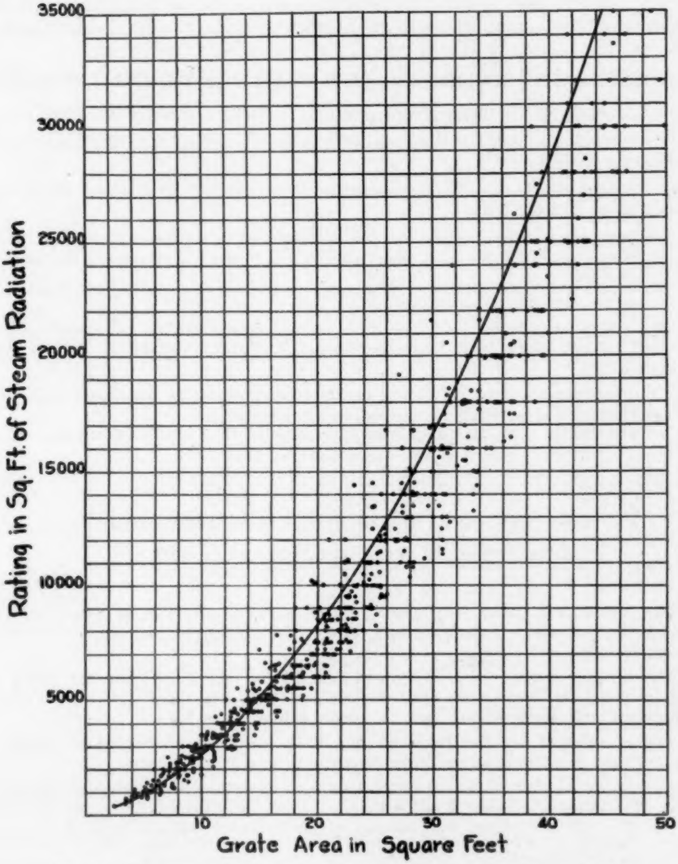

FIG. 2. RELATION OF PRESENT BOILER RATINGS TO GRATE AREA

expressed willingness to be governed by these values with the idea of securing uniformity in the method of rating. The individuality of any boiler manufacturer is not submerged since the question of capacity only is covered and the question of efficiency is left open.

No attempt has been made to evaluate heating surface. Otherwise this Code

might have the aspect of a construction Code rather than a sizing Code. Nevertheless, the requirement of a total heating surface of a certain value will insure a certain minimum efficiency. Efficiencies of a higher degree will be governed entirely by the ingenuity of the manufacturer in providing not only for high furnace efficiency by means of insuring thorough mixing and burning of the gases before they leave the furnace, but also by providing sufficient water volume for the heating surface, thereby preventing steam blocking, with consequent rapid decrease in heat transmission as the rate of heat transfer increases.

CODE FOR THE RATING OF LOW PRESSURE HEATING BOILERS

Adopted by the Steel Heating Boiler Institute December 10, 1929.

1. The purpose of this Code is to provide a uniform method of rating Low Pressure Heating Boilers.

2. The rating of a boiler shall be expressed as square feet of steam or water radiation or as Btu per hour.

3. For purposes of this Code, boilers are divided into two general classes as follows:
   A. Steam and Water Boilers in which solid fuel, hand fired, is used as the heat generating medium.
   B. Steam and Water Boilers in which solid fuel, mechanically fired, oil or gas is used as the heat generating medium.

4. The rating of a boiler in Class *A,* expressed in square feet of steam radiation, shall be not more than fourteen times the heating surface of that boiler in square feet.

5. The grate area of a boiler for the rating as determined by Section 4 shall be not less than that determined by the following formulae:
   For boilers with ratings 300 sq ft to 4000 sq ft of steam radiation;

$$\text{Grate area} = \sqrt{\frac{\text{Catalog Rating (in sq ft Steam Radiation)} - 200}{25.5}}$$

   For boilers with ratings of 4000 sq ft of steam radiation and larger;

$$\text{Grate area} \doteq \sqrt{\frac{\text{Catalog Rating (in sq ft Steam Radiation)} - 1500}{16.8}}$$

6. The rating of a boiler in Class *B,* expressed in square feet of steam radiation, shall be not more than seventeen times the heating surface of that boiler in square feet.

7. The furnace volume of a boiler (as defined in Section 10) for the rating (as determined by Section 6) shall be not less than one cubic foot for every one hundred and forty square feet of steam rating.

8. Boilers selected on the basis of this Code shall be connected to stack and breeching in accordance with the manufacturer's specifications.

9. The rating as defined above for purposes of selection is intended to correspond to the estimated design load, which is to be the sum of items, A, B and C.
   A. The estimated normal heat emission of the connected radiation required to heat the building as determined by accepted practice expressed in square feet of radiation or in Btu per hour.
   B. The estimated maximum heat required by water heaters or other apparatus connected to the boiler, expressed in square feet of radiation or in Btu per hour.
   C. The estimated heat emission of piping connecting radiation and other apparatus to the boiler expressed in square feet of radiation or in Btu per hour.

10. Definitions:
   For purposes of this Code the following definitions will be used:
   A. One square foot of steam radiation shall be considered equal to the emission of 240 Btu per hour and one square foot of water radiation shall be considered equal to the emission of 150 Btu per hour.
   B. Heating surface shall be expressed in square feet and include those surfaces in the

boiler which are exposed to products of combustion on one side and water on the other. The outer surface of tubes shall be used.

C. Grate area shall be considered as the area of the grate surface expressed in square feet and measured in the plane of the top surface of the grate. For double grate boilers the grate area shall be considered as the area of the upper grate plus ¼ the area of the lower grate.

D. Furnace volume shall be considered as the cubical content of the furnace between the top of the base or the normal grate line and the plane of entry into or between the tubes plus the net base volume under the firebox. The net base volume shall be determined by deducting the volume of the refractory lining from the gross volume under the firebox.

The Discussion of this paper appears on page 45 as a Joint Discussion on Report of Continuing Committee on Codes for Testing and Rating Steam Heating Solid Fuel Boilers.

No. 859

AIRATION STUDIES OF GARAGES

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THE problem of heating rather than that of airation or ventilation is the main reason for the acute condition that at times prevails in garages due to the presence of carbon monoxide. This is evident from the investigation which showed that objections to carbon monoxide gas only occur, with very few exceptions, during certain periods of the cold months of the year. During the warm months of the year, practically no trouble is experienced since doors and windows are usually open. Surveys made it apparent that, to an unusual extent, adequate air can be supplied by natural means. As cold weather sets in, more and more of the ventilation openings are closed in order that comfortable temperatures can be maintained within the buildings until, on extremely cold days, very few windows are open, with a result that the carbon monoxide concentration runs high. This condition is particularly troublesome towards the end of the day, possibly due not only to the higher concentrations, but also a decreasing resistance to the effects of CO. The closing of ventilation openings usually can be charged to either an insufficient heating installation or one operated much below capacity for economic reasons.

This investigation was made to determine the use made of airation or natural ventilation as a means of diluting the carbon monoxide concentration of the air in various types of garages.

The research included a great many explorations conducted in four types of garages—Multi-story ramp, Auto service garage, Alley repair shop and the Private home garage, and extended over portions of three years, during both summer and winter conditions. It consisted of a study of the airation, the carbon monoxide concentration of the air within the buildings, and at times the effect of carbon monoxide poisoning upon the employees. In the presentation, each type of garage will be treated separately; such explorations being chosen as will, in the opinion of the authors, best typify the general conditions.

METHOD

Most of the surveys were conducted in the following manner: An airation survey of the building was made to determine the quantity of air entering and leaving. This was done by multiplying the area of each opening by the average

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Presented at the 36th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Philadelphia, Pa., January, 1930.

velocity of air passing through that opening as measured by an anemometer. With such a method, any important discrepancies were easily found because the total inflow of air should approximately equal the total outflow. Samples of the air within the building were taken in at least six localities on each floor, at the floor level, and in the later surveys also at a level about 5 ft above the floor. The samples were taken by purging the original air from 250 cu cm bottles by means of a rubber aspirator bulb with attached scrubber, and were analyzed by the Pyrotannic Acid Method[3] for quantitative determination of carbon monoxide in air. Employees were questioned to determine whether or not they experienced any symptoms of carbon monoxide poisoning.

MULTI-STORY RAMP GARAGES

This type of garage has become popular in congested districts of all large cities for the protected parking and storage of automobiles. It enables the storage or parking of a large number of cars over a relatively small ground area and differs from the ordinary service garage by having a large percentage of the floor space available for parking. Little or no repair work is carried on within the building.

Three such garages were studied: Two are five and eight stories high, respectively, having three walls pierced with openings and the maximum total square feet of openings on each floor less than 3 per cent of the floor area, with the exception of the first floor where there are large door openings. Natural ventilation is employed throughout with the exception of a small fan exhauster, located near the roof on the uppermost floor where cars are washed and the windows are kept closed most of time to maintain comfortable working temperatures.

The buildings are heated by a direct steam radiation and a temperature of not less than 40 F is maintained.

A larger garage of this same type was also studied. It was a 12-story structure, Fig. 1, about 150 ft $\times$ 100 ft with a cubical content of about 1,500,000 cu ft. It has steel windows on all four elevations and is similar to the others in design and layout. It differs as to the heating system and employs unit heaters instead of radiators. The temperature in this building is maintained between 50 and 60 F, in the cold months.

From the garage records, it was found that the average total number of cars handled for one day during the busiest season of the year was about 460 for the five-story garage, 1,050 for the eight-story, and about 1,450 for the 12-story garage. These figures are based upon one day's average of the total number of cars handled during one week not including Sunday. Of the total number of cars handled during any day but Sunday, about 60 per cent enter the buildings between the hours of 8:00 a. m. and 9:30 a. m., and leave between 3:30 p. m. and 5:00 p. m. During these periods, which might be termed the daily rush hours, the CO concentrations varied from one-half of one part to two parts per 10,000 of air, and such high concentrations would not have existed if more windows had been opened, allowing more air to enter and leave the building, and more heat had been supplied to maintain the desired temperature.

One investigation on the 12-story garage showed that about 1,000 cars moved

[3] Technical Paper No. 373, Department of Commerce—The Pyrotannic Acid Method for Quantitative Determination of Carbon Monoxide in Blood and in Air—by R. R. Sayers and W. P. Yant.

in or out of the garage during the rush period of two hours. The average time interval for parking the car or from the starting of the motor until the car leaves the building is about 2 min. The carbon monoxide exhausted was probably 1½ cu ft[4] per car minute. This would mean that about 3,000 cu ft of CO was exhausted into the garage during the two hours, or an average of about 25 cu ft per minute.

The airation survey showed that about 25 per cent of the window openings or the equivalent of less than 1 per cent of the floor area were being utilized or about 50 sq ft for inflow and about the same for outflow per floor. The average velocity of the air passing through the window openings as measured

FIG. 1—THE TWELVE-STORY RAMP GARAGE

by an anemometer was about 250 fpm. The total inflow and outflow was approximately 150,000 cu ft per minute or six air changes per hour. This shows a concentration of about 1 part CO in 6,000 of air or about 1½ parts per 10,000. As soon as the rush period was over the CO was soon diluted to about ½ part per 10,000 or less.

Ordinarily,[5] exposure to concentrations as high as two parts per 10,000 for such a short time would show little, if any, symptoms of CO poisoning. It was found that such a high concentration at the close of a day's work was sometimes just enough to increase the CO concentration in the blood of the employees, who had been exposed to low, ineffective CO concentration all day, to the extent that severe headache resulted.

The suggestion was made that more windows be opened during the rush hours in the afternoon, but this was met with the answer that the inside temperature would be lowered so much that the patrons would protest, thinking

[4] TRANSACTIONS of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, 1921, p. 311.

[5] Physiological Effects of Automobile Exhaust Gas and Standards of Ventilation for Brief Exposures—Yandel Henderson, H. W. Haggard, M. C. Teague, A. L. Prince and Ruth M. Wunderlich. *Journal of Industrial Hygiene*, Vol. 3, July, 1921.

TABLE 1—AIRATION SURVEY OF MULTI-STORY RAMP GARAGE

Average Conditions West Wind—6 Miles Per Hour Inside Temperature—60° Outside Temperature—45°				Survey A	
Floor Number	Average CO in Parts per 10,000 of Air — At Floor	Average CO in Parts per 10,000 of Air — 5 Ft Above Floor	Average Velocity of Inflow fpm	Window Inlet Openings Sq Ft	Inflow Volume cfm
12	0.4	0.4	235	55	12,900
11	0.8	0.8	250	30	7,500
10	0.4	0.4	240	58	13,900
9	0.4	0.4	250	47	11,700
8	0.4	0.4	250	49	12,200
7	0.4	0.4	260	45	11,700
6	0.4	0.4	280	43	12,000
5	0.4	0.4	270	47	12,700
4	0.8	0.8	265	61	16,200
3	0.4	0.4	250	52	13,000
2	Trace	Trace	240	55	13,200
1	Trace	Trace	225	80	18,000
Total	5.0	5.0		622	155,000
Average	0.41	0.41	251	51	12,900

their cars had been exposed to such a temperature all day, thus defeating one of the purposes of protected parking. More heat during the rush hours would, undoubtedly, have eliminated the objectionable situation, since more effective airation would have been possible.

Although the 12-story garage, Fig. 1, handled considerably more cars than either of the other two. Figs. 2 and 3, the CO concentrations were practically the same, since the cubical contents per car in all these garages was not materially different.

The carbon monoxide seemed to be well diffused or distributed. Table 1 shows there was practically no difference in the concentration of carbon monoxide at the floor level and five feet above the floor. This is probably due to the constant agitation of the air caused by the cars being driven to and from the parking stalls. The concentration on the lower floors was always less than on the upper floors due to the large volume of air admitted through the large entrance and exit doors which were being opened and closed continuously. No pockets of high concentration were found.

A typical survey for the summer conditions in one of the five story ramp garages is shown in Table 2.

It will be noted that but a trace of carbon monoxide was found at any time. The number of cars handled in and out was about 200 for the four hours of the survey. The CO exhausted was probably not more than 1 cfm and based on an average of 1½ min running per car, this would mean about 300 cu ft of CO or 75 cu ft per hour, somewhat less than 1½ cfm—with approximately 60,000 cu ft of air per minute blowing through the building this would mean about 0.25 parts per 10,000 of air, which is very hard to measure accurately and is indicated in the table as a trace.

AUTO SERVICE GARAGE

The modern auto service type of garage, Fig. 4, is usually a single or double story structure in which automobiles are both stored and repaired. In the case of the two story garage, the cars are usually stored on the lower floor, while the repair work is normally done on the second floor. The two floors are connected by a ramp or elevator for communication.

In the course of repair work, particularly light repairs and service, the engines exhaust directly into the working zone and contaminate the surrounding air with carbon monoxide gas. During these operations, the car is seldom in motion and the exhaust gases are not so readily diffused as in the Ramp Parking garage.

Observations and tests were made in three garages of this type, one being

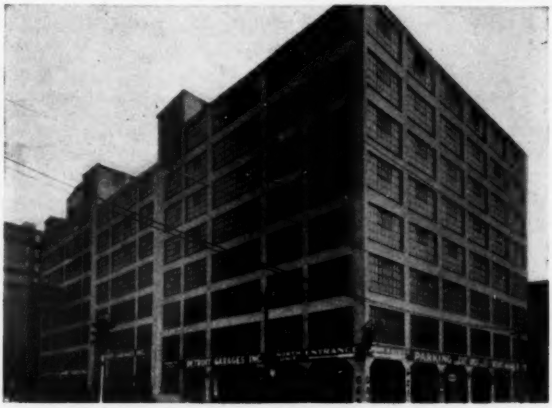

FIG. 2—EIGHT-STORY RAMP GARAGE

a two story structure, Fig. 4, which employed both natural and mechanical means of getting air into and out of the building. Mechanically, air was drawn in through individual ducts, forced through unit heaters and exhausted at about 500 fpm velocity near the ceiling. These heaters were provided with a damper which could be adjusted to provide total or partial recirculation of inside air. A mechanical exhaust duct system in the floor exhausted about 50 per cent of the total inlet capacity of unit heaters while open windows provided the remaining necessary exhaust outlets, and, at the same time, additional inlets.

This building, Fig. 4, has a content of about 500,000 cu ft. Steel windows are used in all four elevations. The total ventilating window area available is about 2,000 sq ft or about 4 per cent of the floor area.

Tests were conducted in this garage during the months of January and February, a time when repair work was not at a peak, although the average amount was being done for that particular time of year. Most of the tests were made with the unit heaters set for total recirculation just as they had

been operating probably ever since installation and, as a result, a great many windows were found open to provide inlet as well as outlet of air.

One survey showed that about 35 cars moved in and out during the day for all types of repairs from minor adjustments to major repairs. As nearly as could be observed, the average time that the engine was running was about 8 min. This would mean about 280 total car minutes, and based on 1½ cu ft of CO being exhausted per minute per car, there would be exhausted a total of 420 cu ft of CO. There were about 95 sq ft of windows serving as inlets for air, and a similar amount for outflow, or a total of approximately 0.5 per cent of the floor area. The average inflow velocity was about 180 fpm, giving an inflow and outflow of about 18,000 cfm or about 8,000,000 cu ft per eight hour day, two air changes per hour. This volume also, includes the inflow through the entrance door. Under such conditions, the carbon monoxide concentration ranged from a trace to about one-half of one part per 10,000 of air, which, of course, registered no ill effects upon the workmen. On this particular occasion, the inside temperature was about 60 deg and the outside temperature averaged 25 deg, survey C—Table 3. The unit heaters were operated on a 100 per cent recirculating basis.

During a rush period when 50 or more cars are handled each day, and the same volume of air flowing through the building, the average carbon monoxide concentration would be about one part per 10,000 and at times possibly two parts per 10,000 or more. Under such conditions, if the weather was cold, and less windows were open the conditions would be acute. This is particularly true when the men have been exposed to, say, one-half part of CO per 10,000 of air during most of the day and, due to an increase in the number of cars moved near the end of the day, the CO concentration is sometimes raised to two parts or more per 10,000 for a short time. The answer to this is that in accordance with the volume of CO exhausted from running autos, sufficient air must be introduced to dilute the CO concentration below a disagreeable degree and whether this required air is introduced through open windows, or otherwise, it must be heated to a comfortable temperature.

TABLE 2—AIRATION SURVEY OF FIVE-STORY RAMP GARAGE

Average Conditions S. W. Wind—6 Miles Per Hour			Survey B		
Time A. M.	Floor Number	Aver. CO in Parts Per 10,000 of Air	Aver. Velocity of Inflow	Window Inlet Openings Sq Ft	Inflow Volume cfm
9:00	5	Trace	307	46	14,000
10:00	4	Trace	328	50	16,500
11:00	3	Trace	333	40	13,300
12:00	2	Trace	305	52	16,000
				188	59,800
Average per floor		Trace	318	47	14,900

The foregoing statements are made on the basis that no direct method of exhausting the fumes from the cars is used to take the gases through some exhaust duct system to the outside.

Tests were made on another day under almost identical conditions, with the exception that no air was recirculated through the heaters. Under these conditions, it was found that about 20,000 cu ft of air per minute was passing through the building and the carbon monoxide concentration was the same as before—ranging from a trace to about one-half of one part per 10,000 of air. The outside temperature was 17 deg and the inside temperature was about 56 deg, survey D—Table 3.

In either case, only a portion of the total available ventilation was being used, but if more windows had been opened under the conditions, without

FIG. 3—TYPICAL FIVE-STORY RAMP GARAGE

supplying more heat, the inside temperature would, undoubtedly, have been lowered to an uncomfortable degree. This was proven out by other tests made on days when the outside temperature was very close to zero, under which conditions the inside temperature varied between 48 and 52 deg. Such temperatures were uncomfortable, but the workmen were content to sacrifice comfort for air low in CO concentration.

In general, it can be said that the carbon monoxide was well diffused; that is, the concentration at the floor level and 5 ft above the floor was practically the same, and this seemed to be true whether the air was admitted through the window openings as natural ventilation or through the unit heaters as mechanical ventilation. Probably it was the agitation of air caused by the high velocity of the unit heater exhausts, in either case, which caused such a complete diffusion of the carbon monoxide gas.

This garage was cited as one in which considerable trouble was experienced from high concentrations of carbon monoxide gas but results of tests conducted on 14 different days, when conditions were most favorable for high CO con-

TABLE 3—RESULTS OF TWO AIRATION SURVEYS OF A TWO-STORY AUTO SERVICE TYPE GARAGE

Average Conditions—Survey C				Average Conditions—Survey D		
West Wind—4 Miles Per Hour Inside Temperature—62° Outside Temperature—25°				Northwest Wind—6 Miles Per Hour Inside Temperature—56° Outside Temperature 17°		
Survey Number	Floor Number	Aver. CO in Parts Per 10,000 of Air		Aver. Velocity of Inflow fpm	Inlet Openings sq ft	Inflow Volume cfm
		At Floor	5 Ft Above Floor			
*C	1	0.4	0.4	180	95	18,000
	2	0.5	0.5			
**D	1	0.5	0.5	500	33-U.H	20,000
	2	0.5	0.5	150	20-W	

* During Survey C air was introduced through windows only.
** During Survey D air was introduced through unit heaters and window openings. The 500 velocity was through the heaters and the 150 was the velocity through 20 sq ft of window opening.

centrations, seemed to indicate that there was more discomfort experienced from inadequate heating than from carbon monoxide poisoning.

Two other garages of the auto service type were studied, but not so extensively as the one just discussed—these are of the common single story type, having three walls pierced with ventilation openings whose total area is about 3 per cent of the floor area. Natural ventilation is employed throughout, and heat is supplied by direct steam radiation.

As a result of the few tests conducted in these buildings, and information gathered from the management, it was found that as long as the outside temperatures permitted the opening of at least 50 per cent of the total window ventilators, practically no symptoms of carbon monoxide poisoning were experienced.

Tests made during near zero weather showed that the CO was not very uniformly diffused throughout the building. Air samples taken at the floor level showed slightly higher concentrations than those taken about 5 ft above the floor, and the highest concentration recorded was between 1½ and 2 parts per 10,000 of air. This condition existed directly after a motor tuning operation, but only for a few minutes until the doors were opened to clear the building of the smoke and gas, or opened to let a car in or out. During such cold weather the windows and doors are usually kept closed, but as soon as a motor is run for tuning or other adjustments, the doors are opened until the operation is completed. In other words, the workmen have learned their lesson; they prefer to dilute the poisonous gas before it has time to register any ill effects upon them, but in doing so, they usually sacrifice comfortable working temperatures, at least temporarily.

ALLEY REPAIR SHOPS

Garages of this type are small auto repair shops usually found up an alley and located in an old barn or storage shed or some other structure which was never designed to serve such a purpose. Usually these shops can only

accommodate from four to six cars and are provided with no other ventilation openings except the service doors and possibly a window or two.

A few of the shops studied are heated by direct steam radiation, others by stoves and a few are not provided with heating equipment of any kind. Where no heating equipment is provided, practically no work is done during severe cold weather.

As a result of observations made in a number of these repair shops, and of information gained from questioning employees and the proprietors, it was found that, on the average, carbon monoxide poisoning is not as serious a problem in these shops as it is in the Service Garages. The reason for this is that a motor is seldom run within the building unless the doors are open.

FIG. 4—THE TWO-STORY AUTO SERVICE GARAGE

Practically all motor testing and running-in of bearings is done outside. It is interesting to note that one was provided with an old flexible metal hose for piping the exhaust to the outside.

PRIVATE HOME GARAGE

A few tests were conducted in a private, two-car garage to determine the rate of increase of the carbon monoxide concentration of the air for the conditions with the engine idling and the engine racing, while all doors and windows of the garage were tightly closed.

A four cylinder Chevrolet car was placed in a double garage, and after all doors and windows were closed, the motor was started and allowed to run at an idling speed for 10 min. Samples of the air at floor level and breathing level were taken every 2 min. At the end of the 10-min period, the motor was stopped, the doors were opened and the garage was allowed to air for about 5 min. Then the doors were again closed, the throttle on the motor was set at a racing speed and the same sampling procedure was followed as before.

On the basis that the garage contained about 3,000 cu ft of air, it is seen that with the engine exhausting between 1 and 1½ cu ft of carbon monoxide per

minute, the CO concentration would be about four to five parts per 10,000 for the first minute of running, which, of course, would be a dangerous concentration to breathe. After 15 min running, the concentration would be between 50 and 60 parts per 10,000 which would probably be fatal. Knowing this to be the case, the survey was made by going in and out through an outside door every 2 min while air samples were being taken. Owing probably to the fact that air was introduced during each entrance and exit, the samples were not indicative of the true condition and, since no reasonable check on the data was obtained, the results of these tests are not shown. Certainly all indications pointed to the fact that a small, closed garage is no place for a man to be if the engine is running.

CONCLUSIONS

Multi-story Ramp Garage

1. Sufficient air can be furnished through window openings, as natural ventilation, to keep the carbon monoxide concentration below a harmful or even disagreeable degree.

2. Higher concentrations of carbon monoxide were found to occur on extremely cold days when it became necessary to close all or all but a few windows in order that comfortable inside temperatures might be maintained.

3. Tests showed that the carbon monoxide content of the air in this type of garage is practically the same at the floor level as it is five feet above the floor. This is due to the fact that a motor is seldom running unless the car is in motion and the gas is well diffused by the action of cars going up and down the ramps.

4. Rush periods occur between the hours 8:00 a.m. to 9:30 a.m. and 4:00 p.m. to 5:30 p.m. during which periods 60 per cent of the total cars handled throughout the day enter and leave the buildings respectively. The CO concentrations during these periods were found to range from one-half to two parts per 10,000 of air, depending upon the amount of air movement.

5. The carbon monoxide concentration between the rush hours was almost always below one-half part per 10,000 of air which produced no ill effects.

6. The CO concentration during the rush hour was sometimes high enough to increase the existing low concentration in the blood of the attendants to the extent that headache developed.

7. The total available ventilation area was never utilized except on hot days in the summer.

Auto Service Garage

1. Practically no trouble would be experienced from carbon monoxide poisoning in the garages studied if some reasonable percentage of the available ventilation areas were utilized.

2. Where unit heaters are employed, the carbon monoxide is more thoroughly diffused than where direct radiation is used.

3. In the garages studied, sufficient heat was not supplied in cold weather to warm the necessary volume of air to a comfortable temperature.

Alley Repair Shops

1. Carbon monoxide poisoning is not a serious problem in these shops, since motors are seldom operated within the building unless the doors are open.

Private Home Garages

1. A four cylinder motor operating at a racing speed in a two car garage with doors and windows closed would produce a concentration of carbon monoxide which would probably cause death to anyone exposed to it for 10 min or more.

DISCUSSION

A. VOGEL (WRITTEN): The problems of natural ventilation are of great importance in large industrial buildings. Industrial plants of today are changing in character very rapidly. In fact, it is freely predicted that the industrial plant of the near future will consist generally of one large building. We have already prepared studies of two plants for the General Electric Co., which will be essentially single buildings of such large area that the problems of natural ventilation will be very serious.

In the *Illuminating Engineering Society* we have had a Committee on Natural Lighting for some years, and the work of that committee has progressed to the extent that we are able to predict quite accurately the natural lighting in buildings. In fact, in one plant recently completed in Peterboro, Ontario, Canada, the natural ilumination is of such excellent quality that I doubt whether it can be improved economically.

We hope that the Natural Ventilation Committee of the Society will be able to do something comparable in the natural ventilation of buildings.

Let us consider as an example one of the new type plants—a building of large area approximately 900 ft by 1,800 ft. The only means of natural ventilation, of course, will be through the roof and side walls. The side walls, however, are at such great distances from the central portion of the building that they must be largely neglected. It seems, then, that we must consider roof ventilation only. One of our members, Mr. Randall, has carried out quite a number of experiments in the natural ventilation of such buildings and has found that the prevailing winds cause air to enter certain skylights, pass through the building, and leave by other skylights. Sufficient experiments have been performed so that this form of natural ventilation can be determined from the plans of the building and Weather Bureau data.

It has been found, also, that when large volumes of heat are generated in manufacturing processes, a very appreciable effect is produced on the natural ventilation of the building. It will be necessary to make measurements and study this problem over a period of several years before any conclusion can be drawn.

The work of Professor Emswiler in natural ventilation is also noteworthy and so it may be observed that our Committee has started with some basis for its work, but we need the cooperation of many members of this Society in order to carry it forward.

Referring again to the *Illuminating Engineering Society* we started with a very small number in the Natural Lighting Committee, and now that committee has reached a membership of nearly 20 members and includes men interested in

illumination from all over the United States. It is hoped that the Committee on Natural Ventilation of this Society will grow and that we will have men interested in this subject from all over, including not only men interested in industrial plants, as I am, but also men interested in the design of large office buildings, such as the recently completed Chrysler Building in New York, and also other buildings wherein natural ventilation has a decided effect on the heating system.

The particular problem Mr. Randall presented, relating to the airation of garages, is of interest not only in connection with garages but also in industrial plants where fumes of various types are given off. Most states have codes requiring that fumes from plating rooms, etc., be exhausted by mechanical means, but the codes generally do not mention oil furnaces welding, and other processes. The solution of all these problems can be very much facilitated by a thorough study of natural ventilation and the members cooperation with the Committee on Natural Ventilation is earnestly desired.

E. K. CAMPBELL: I want to compliment Mr. Randall very highly for this work. He and his associates have done an immense amount of painstaking work and have reached certain conclusions which, I think, are binding. It is unfortunate, however, that due in part, at least, to improper analysis of the complete problem, or possibly because it was approached from a different angle, they did not investigate or take many tests during what we might call the peak load periods in the garage. It is essential to know conditions during the arrival of the greatest number of cars, while they are being put away also at times when a great number are warming up to go out.

I think the author's conclusion that the CO problem in a garage is directly related to the heating problem is correct. I also think that it applies to a lot of other things and is the important consideration in settling the open window ventilation problem. Open window ventilation is good under most conditions as long as the windows can be opened with comfort. When comfort disappears ventilation problem goes with it. The CO problem begins when windows are closed for lack of comfort and under peak load operation.

W. H. SCHULZE: The Baltimore City Health Department has just recently started some work or is intending to start some work on the concentration of carbon monoxide in garages. We have done a lot of work on carbon monoxide as given off from gas appliances and the City of Baltimore has in force a gas appliance ordinance which takes care of the registration of gas appliances. We intend to go into this garage work more as a study of the effects of small doses of carbon monoxide rather than large doses. In other words, a chronic carbon monoxide poisoning. I am greatly interested to hear that a good deal of this work has been done in garages, but I am sure that a great deal remains to be done.

W. C. RANDALL: With reference to the remarks of Mr. Campbell, on this question of peak loads, manifestly our investigation was limited at that point. With few exceptions our investigations indicated maximums around two and a half to three parts per 10,000. So far as the Society is concerned and so far as any code is concerned, I am quite sure that further investigation should be made particularly with reference to those conditions in which a more acute situation prevails than those considered by the authors.

I would call Mr. Schulze's attention to the references in the paper dealing with some very commendable work along the lines he mentioned. The authors

felt that they could accept the criteria of some place between a foot and a foot and a half carbon monoxide per minute running time of the car and we could accept some of the figures as to the danger point and, incidentally, the so-called grumble point, as it might be called around two to two and a half, when the garage men begin to kick.

With reference to the work of the Natural Ventilation Committee that Mr. Vogel has referred to, I have had the pleasure of working with Mr. Vogel on day lighting committees. I think we have accomplished considerably more on day lighting than we have on natural ventilation.

As to the method of predicting the flow, due to the forces of nature, temperature and wind and a combination thereof, I have collected some data, and at some future time I have in mind presenting to the Society for their approval some very practical methods not entirely accurate scientifically, but for all intents and purposes certainly a lot closer and a lot better than the general methods used today.

H. M. HART: It would seem to me that inasmuch as the committee are pretty well agreed on the amount of carbon monoxide generated per car, the real need now is to conduct some investigation on the number of cars in operation in garages of different types at the peak load. That is the problem that always comes up to the engineer when he is trying to design a ventilating system to take care of a garage. An ordinance in Chicago requires that there shall be provided 5,000 cu ft of air per minute per car under operation in a garage, but the question is, how many cars are going to be in operation? The only way they have of getting around that is to have a written agreement with the owner of the garage on certain limits, but I doubt whether the specified limits are adhered to.

E. C. EVANS: All of the work that the committee has been doing has been on garages above ground. I would just like to offer a few suggestions that in Pittsburgh we have a very modern, up-to-date office building in which there are five floors of garage space below the ground. I am quite sure the committee could get permission to investigate perhaps not for publicity but just to have a garage below ground that they could test and know what takes place in a space of that type.

THREE-STORY EXPERIMENTAL INSTALLATION IN THE INTERNAL COMBUSTION ENGINE LABORATORY AT THE AGRICULTURAL AND MECHANICAL COLLEGE OF TEXAS

No. 860

PIPE AND ORIFICE SIZES FOR SMALL GRAVITY CIRCULATION HOT WATER HEATING SYSTEMS

By ELMER G. SMITH[1] (*MEMBER*), COLLEGE STATION, TEXAS

The results of co-operative research between the A. S. H. V. E. and the Texas Engineering Experiment Station, Dr. F. E. Giesecke, Director

WHEN a hot water heating system is installed in a small building, such as a residence, the two-pipe underfoot feed, direct return system with bare mains is usually chosen because it is cheaper to install than the reversed return system, gives smaller heat losses in the basement, and the piping looks less clumsy. In the past, some trouble has been experienced with the circulation in direct return systems, but recent experiments at this station have demonstrated that when orifices of the proper size are placed in each radiator circuit, the circulation becomes not merely satisfactory but remarkably good. The usual place for such an orifice is in the union at one end of the radiator. There are at least three ways of getting these orifices into the circuit:

1. The contractor may make them out of sheet copper and insert them into the unions as illustrated by Figs. 1 and 2. The copper should be about 0.025 in. thick (22 gage).
2. The contractor may specify valves equipped with orifices of various sizes. There is at least one such valve now on the market. This valve is intended for use with vapor systems, but may also be used with hot water systems.
3. The contractor may specify valves that have adjustable calibrated orifices. There are at least two such valves on the market both of which are intended for use with vapor systems.

In order to make the installations of these systems easy and practical the following tables have been prepared from the results of experimental work:

Table 1, giving the size of riser required to carry a given number of square feet of radiation.

Table 2, giving the size of mains required to carry a given number of square feet of radiation for various lengths of line when the mains are between 3 and 4 ft above the mid-point of the heater.

Table 3, giving the size of mains required to carry a given number of square

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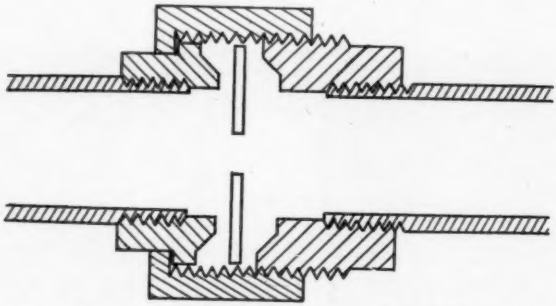

FIG. 1. ORIFICE AS PLACED IN UNION BEFORE UNION IS TIGHTENED

feet of radiation when the mains are between 4 and 5 ft above the mid-point of the heater.

Table 4, giving the size of main heater risers required to carry a given number of square feet of radiation.

Table 5, giving the diameters of the orifices that should be used with various sizes of radiators for different heights above the mid-point of the heater. This table is to be used when the contractor makes his own orifices.

Table 6, giving the sizes of the orifices for use with valves that come equipped with them.

Table 7, giving the proper settings for use with valves with adjustable calibrated orifices.

EXPERIMENTAL WORK

The first experiment on a direct return system was made before the tables were calculated. This plant had seven radiators all between 30 sq ft and

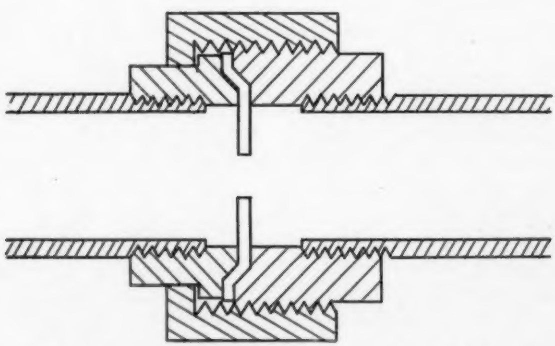

FIG. 2. ORIFICE AS PLACED IN UNION AFTER UNION IS TIGHTENED

70 sq ft and represented one half of a two-story installation. This system was very successful and the results were fully described in a previous paper entitled "Pipe Sizes for Hot Water Heating Systems," by F. E. Giesecke and the writer which appeared in the A.S.H.V.E. TRANSACTIONS 1929. After a series of tests on this system and also on a reversed return system, the theory was verified sufficiently to justify an attempt to calculate the tables.

In their present form the tables are not designed to produce the best possible circulation, but represent rather the simplest tables that could be produced and still be assured of reasonably good circulation. When the tables had finally taken shape the system shown in Fig. 11, page 278, was designed in accordance with them. When the system was actually built it was necessary to bend the mains back upon themselves in order to get so large a system into the space that was available. Also, the only heater available was a converter intended to heat water for shower baths. The openings in this heater were only 3 in. instead of 3½ in. or 4 in., as called for by the tables. In addition to this the vertical distance from the mid-point of the heater to the mid-point of the mains directly above was made 4 ft, the minimum distance for which Table 3 can be used.

Both mains were bare throughout their entire length and the fact that they were not covered by a tight floor probably tended to cause the radiators near the end of the line to receive cooler water than they otherwise would have.

The results of this test are shown in the table on the following page.

In order to more readily visualize the meaning of these figures, it will be helpful to divide the radiators into two groups (1) medium-sized radiators having more than 35 sq ft of surface and (2) small-sized radiators having less than 16 sq ft of surface.

It will be noted that the temperatures in the medium-sized radiators are very closely grouped at about 176 F. The hottest is No. 31 (181 F), and the coldest is No. 32 (168.5 F), and the temperature difference between these extremes is only 13½ F, making the deviation from the mean only 6¼ F which is

Group 1—Medium-Sized Radiators			Group 2—Small-Sized Radiators		
Radiator Number	Surface in Sq Ft	Average Temp. of Water—F	Radiator Number	Surface in Sq Ft	Average Temp. of Water—F
11	74¼	173			
12	38½	179.7			
			13	13¾	166
14	74¼	177			
			15	13¾	164
21	74¼	177.7			
22	38½	175			
23	74¼	177			
24	74¼	174			
			25	13¾	166
31	75	181			
32	55	168.5			
33	75	175			
			34	15	163
			35	15	161

RECORD OF TESTS CONDUCTED ON EXPERIMENTAL PLANT BUILT ACCORDING TO TABLES 1, 3, 4, AND 6.

TIME REQUIRED FOR WATER TO REACH LAST RISER, 17 MIN.

READINGS TAKEN 1 HOUR AFTER STARTING HEATER

Radiator Number*	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Temperature of Water Entering the Radiator	178	177	172	163	159	173	169	171	164	156	172	163	166	152	145
Temperature of Water Leaving the Radiator	132	140	115	140	132	138	139	141	137	132	141	138	139	129	125
Average Temperature of Water in the Radiator	150	158.5	143.5	152.5	145.5	155.5	154	156	150.5	144	156.5	150.5	152.5	140.5	135

READINGS TAKEN 4½ HOURS AFTER STARTING HEATER

Radiator Number*	11	12	13	14	15	21	22	23	24	25	31	32	33	34	35
Temperature of Water Entering the Radiator	200	199	193	189	184	198	191	194	186	177	198	185	192	170	165
Temperature of Water Leaving the Radiator	146	160.5	135	165	148	157.5	159	160	162	155	164	152	158	146	144
Average Temperature of Water in the Radiator	173	179.7	164	177	166	177.7	175	177	174	166	181	168.5	175	158	154.5

* For numbering of radiators see page 277.

Table 1. Riser Sizes*

Pipe-Size—In.		Capacity in Sq Ft	
Flow	Return		
½	½	0.0	15.4
½	¾	15.4	21.0
½	1	21.0	23.5
¾	¾	23.5	33.2
¾	1	33.2	43.6
¾	1¼	43.6	47.0
1	1	47.0	63.2
1	1¼	63.2	81.0
1	1½	81.0	87.6
1¼	1¼	87.6	130
1¼	1½	130	157
1½	1½	157	196
1½	2	196	251
1½	2½	251	270
2	2	270	384

*All radiators less than 12 ft above the mid-point of the heater must be connected to the mains by independent risers.

unusually small. These two radiators happen to be next to each other on the second floor and in the half of the line nearer the heater. This seems to indicate that such differences as do exist are caused, not by the position on the line, but rather by the impossibility of selecting exactly the correct resistance for any given radiator, due to the fact that the resistances of both the risers and the orifices vary by jumps.

Referring again to radiators No. 31 and No. 32, will be noticed by reference to Table 6 that the (31-40) orifice called for by radiator No. 31 has a range of 72 sq ft to 90.4 sq ft. Radiator No. 31 is near the low end of this range. Its temperature should therefore be a little high. Similarly, the (21-30) orifice required by radiator No. 32 has a range of 43.9 sq ft to 58.5 sq ft. Radiator No. 32 has 55 sq ft of surface. It is therefore near the upper end of this range and its average temperature should therefore be a little low.

Referring to the very small radiators, it is evident that although these radiators agree closely among themselves, they are on the average about 13 F cooler than the larger radiators and that the coldest radiator in this group is a full 20 F cooler than the hottest one in the larger group. This represents a variation of 10 F from the mean. Small radiators dissipate heat more efficiently than larger ones, but even when that is taken into consideration the small radiators are 10 per cent to 15 per cent below the larger ones in the number of Btu per square foot that they dissipate. As a rule this difference would probably not be noticed by the occupants of the building, but it may be advisable to increase the surface of very small radiators by about 10 per cent to obviate any possibility of having trouble.

How to Select Sizes of Pipes and Orifices

When the various tables are applied in every-day work it will probably be remarked that when mixed pipe sizes are used the flow main or riser, as the case may be, is always smaller than the return main or riser. There are two reasons for this: First, the heat losses due to cooling in the pipes are less

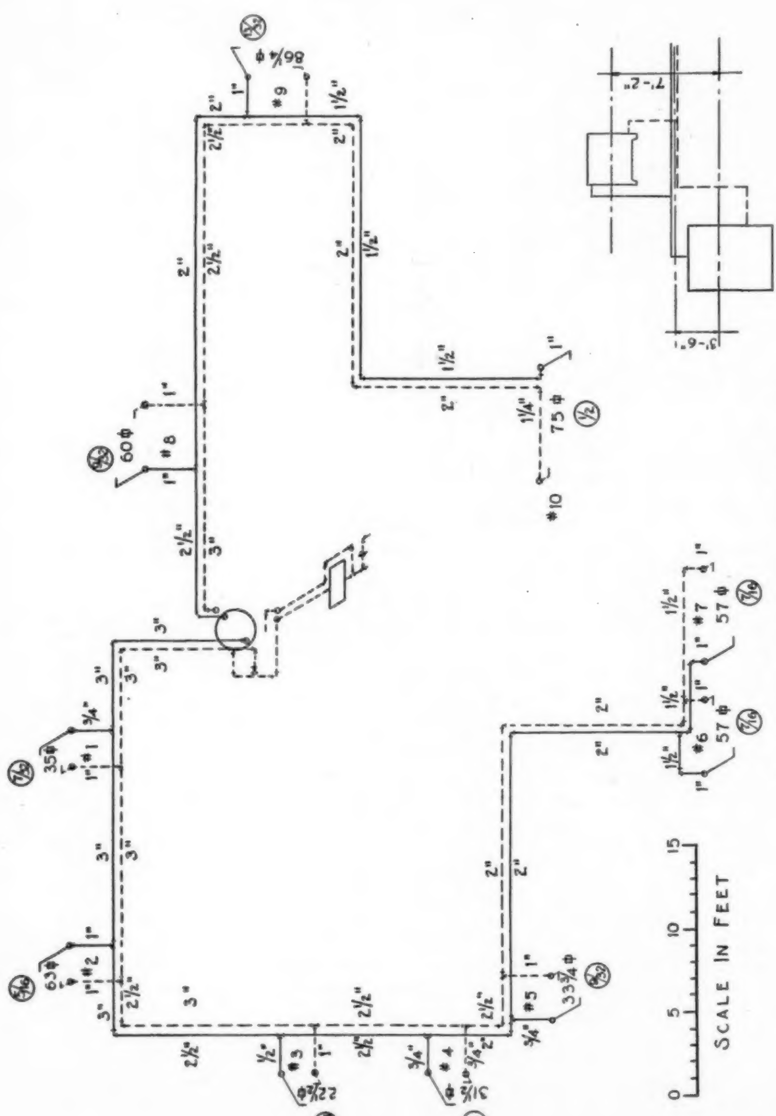

FIG. 3. LAY-OUT FOR ONE-STORY RESIDENCE
The sizes of the orifices are indicated by a circle with a number inside.

with this arrangement than when the flow main or riser is larger; and second, it helps the circulation slightly because cooling in a flow riser lowers the average temperature of a radiator, whereas, cooling in a return riser simply produces more rapid circulation of the water.

In the case of the risers it will also be noticed that the return riser is sometimes two sizes larger than the flow riser as, for instance, a ¾ in. flow riser may be used with a 1¼ in. return riser. This is done to help equalize the friction in the risers.

ONE-STORY RESIDENCE PROBLEM

The simplest kind of plant to install is one intended to heat a one-story residence. Such a plant is illustrated in Fig. 3. Before proceeding with the selection of the pipe sizes it may be well to consider some of the piping details. The risers may be connected to the mains as illustrated in Fig 4A, Fig. 4B or Fig. 4C, page 256. Since Fig. 4B is already in common use, it will be selected for this problem. The radiators may be connected with a low flow connection illustrated in Fig. 5A, page 262, or with a high flow connection as illustrated in Fig. 5B. The high flow connection locates the valve more conveniently and heats more efficiently. This connection will therefore be selected. In order to be able to refer to them conveniently, the radiators have been numbered consecutively beginning at the heater. The left-hand branch begins with radiator No. 1, and the right-hand branch with radiator No. 8.

HOW TO SELECT RISER SIZES

By reference to Table 1, page 251, it will be noted that the riser sizes should be as follows:

	Flow—In.	Return—In.
Radiator No. 1 (35 sq ft)	¾	1
Radiator No. 2 (63 sq ft)	1	1
Radiator No. 3 (22½ sq ft)	½	1
Radiator No. 4 (31½ sq ft)	¾	¾
Radiator No. 5 (33¾ sq ft)	¾	1
Radiator No. 6 (57 sq ft)	1	1
Radiator No. 7 (57 sq ft)	1	1
Radiator No. 8 (60 sq ft)	1	1
Radiator No. 9 (86¼ sq ft)	1	1½
Radiator No. 10 (75 sq ft)	1	1¼

HOW TO SELECT SIZES FOR MAINS

There are two tables of pipe sizes for mains: Table 2 is to be used when the distance from the mid-point of the heater to the mid-point of the lowest portion of the mains is between 3 and 4 ft and Table 3 for use when this distance lies between 4 and 5 ft. The mid-point of a heater is a point midway between the flow connection and the return connection, not a point midway between the top of the heater and the floor, as might be supposed. In the case of this particular plant the distance from the mid-point of the heater to the mid-point of the mains is 3 ft 6 in. Table 2, therefore, must be used.

In order to select the pipe sizes for a given section of the mains, the length

TABLE 2. CAPACITIES OF MAINS IN SQUARE FEET OF DIRECT RADIATION (DISTANCE FROM MID-POINT OF HEATER TO LOWER PORTIONS OF THE MAINS 3 FT-4 FT)

Pipe Sizes		Length of Line in Feet																				Pipe Sizes	
Flow	Return	0	20	20	25	25	30	30	35	35	40	40	45	45	50	50	60	60	70	70	80	Flow	Return
¾	¾	20	23	0	20	30	18	0	17	0	16	0	15	0	15	0	14	0	13	0	12	¾	¾
¾	1	23	30	20	26	18	24	17	22	16	21	15	20	15	20	14	18	13	17	12	15	¾	1
1	1	30	43	26	39	24	36	22	33	21	31	20	29	20	28	18	25	17	23	15	21	1	1
1	1¼	43	57	39	53	36	49	33	45	31	41	29	40	28	39	25	34	23	32	21	34	1	1¼
1¼	1¼	57	83	53	80	49	75	45	70	41	67	40	62	39	59	34	54	32	51	34	47	1¼	1¼
1¼	1½	83	101	80	96	75	89	70	84	67	80	62	75	59	71	54	64	51	61	47	56	1¼	1½
1½	1½	101	130	96	120	89	110	84	105	80	100	75	95	71	90	64	80	61	75	56	70	1½	1½
1½	2	130	169	120	155	110	135	105	136	100	129	95	122	90	117	80	106	75	99	70	92	1½	2
2	2	169	240	155	220	135	205	136	195	129	180	122	170	117	165	106	155	99	144	92	135	2	2
2	2½	240	285	220	264	205	240	195	235	180	220	170	195	165	200	155	187	144	175	135	165	2	2½
2½	2½	285	350	264	330	240	315	235	295	220	280	195	270	200	255	187	235	175	220	165	210	2½	2½
2½	3	350	450	330	442	315	400	295	377	280	360	270	345	255	330	235	300	220	285	210	272	2½	3
3	3	450	630	442	585	400	545	377	520	360	500	345	475	330	460	300	430	285	400	272	385	3	3
3	3½	630	730	585	683	545	640	520	610	500	585	475	558	460	540	430	510	400	470	385	450	3	3½
3½	3½	730	870	683	820	640	780	610	740	585	705	558	680	540	660	510	620	470	580	450	550	3½	3½
3½	4	870	997	820	925	780	880	740	840	705	800	680	770	660	740	620	700	580	670	550	622	3½	4
4	4	997	1170	925	1080	880	1020	840	970	800	930	770	890	740	850	700	800	670	760	622	720	4	4
4	5	.....	.....	1080	1370	1020	1290	970	1230	930	1190	890	1140	850	1100	800	1030	760	980	720	925	4	5
5	5	.....	.....	1370	1860	1290	1770	1230	1700	1190	1650	1140	1600	1100	1540	1030	1460	980	1380	925	1300	5	5
5	6	.....	.....	1860	2250	1770	2170	1700	2080	1650	1960	1600	1970	1540	1900	1460	1800	1380	1700	1300	1600	5	6
6	6	.....	.....	2250	2840	2170	2800	2080	2900	1960	2650	1970	2560	1900	2490	1800	2340	1700	2203	1600	2100	6	6
		80	90	90	100	100	120	120	140	140	160	160	180	180	200	200	240	240	280				
¾	¾	0	11	0	10	0	9	0	8	0	7	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾	¾
¾	1	11	14	10	13	9	12	8	10	7	9	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾	1
1	1	14	20	13	19	12	17	10	15	9	14	0	13	0	12	0	11	0	10	.....	.....	1	1
1	1¼	20	28	19	26	17	23	15	21	14	20	13	18	12	17	11	16	10	15	.....	.....	1	1¼
1¼	1¼	28	44	26	41	23	38	21	34	20	32	18	30	17	29	16	29	15	27	.....	.....	1¼	1¼
1¼	1½	44	54	41	49	38	46	34	41	32	39	30	36	29	35	29	34	27	31	.....	.....	1¼	1½
1½	1½	54	65	49	60	46	58	41	52	39	50	36	45	35	45	34	40	31	35	.....	.....	1½	1½
1½	2	65	87	60	81	58	76	52	69	50	67	45	61	45	60	40	53	35	48	.....	.....	1½	2
2	2	87	130	81	125	76	110	69	105	67	100	61	95	60	90	53	80	48	75	.....	.....	2	2
2	2½	130	153	125	150	110	136	105	127	100	118	95	110	90	103	80	93	78	88	.....	.....	2	2½
2½	2½	153	185	150	190	136	180	127	160	118	145	110	130	103	120	93	110	88	105	.....	.....	2½	2½
2½	3	185	247	190	240	180	235	160	214	145	189	130	171	120	160	110	146	105	138	.....	.....	2½	3
3	3	247	370	240	350	235	340	214	320	189	270	171	250	160	240	146	215	138	200	.....	.....	3	3
3	3½	370	434	350	410	340	390	320	367	270	322	250	300	240	286	215	254	200	237	.....	.....	3	3½
3½	3½	434	525	410	500	390	460	367	430	322	400	300	370	286	350	254	310	237	200	.....	.....	3½	3½
3½	4	525	596	500	570	460	524	430	490	400	460	370	438	350	410	310	368	290	344	.....	.....	3½	4
4	4	596	690	570	600	524	610	490	570	460	540	438	510	410	490	368	450	344	420	.....	.....	4	4
4	5	690	890	600	850	610	790	570	738	540	700	510	668	490	644	450	600	420	560	.....	.....	4	5
5	5	890	1260	850	1200	790	1120	738	1050	700	1000	668	970	644	940	600	890	560	850	.....	.....	5	5
5	6	1260	1550	1200	1488	1120	1400	1050	1315	1000	1255	970	1226	940	1160	890	1120	850	990	.....	.....	5	6
6	6	1550	2000	1488	1950	1400	1850	1315	1760	1255	1680	1226	1620	1160	1580	1120	1520	990	1470	.....	.....	6	6

TABLE 3. CAPACITIES OF MAINS IN SQUARE FEET OF DIRECT RADIATION (DISTANCE FROM MID-POINT OF HEATER TO MID-POINT OF MAIN DIRECTLY ABOVE 4 FT-5 FT)

Pipe Sizes		Length of Line in Feet																				Pipe Sizes	
Flow	Return	0	20	20	25	25	30	30	35	35	40	40	45	45	50	50	60	60	70	70	80	Flow	Return
¾	¾	0	27	0	23	0	21	0	20	0	18	0	17	0	17	0	16	0	15	0	14	¾	¾
¾	1	27	35	23	30	21	28	20	25	18	24	17	23	17	23	16	21	15	20	14	17	¾	1
1	1	35	50	30	45	28	41	25	38	24	36	23	33	23	32	21	29	20	27	17	24	1	1
1	1¼	50	66	45	61	41	57	38	52	36	47	33	46	32	44	29	39	27	37	24	39	1	1¼
1¼	1¼	66	96	61	92	57	87	52	81	47	77	46	71	44	68	39	62	37	59	39	54	1¼	1¼
1¼	1½	96	117	92	111	87	103	81	97	77	92	71	87	68	82	62	74	59	70	54	65	1¼	1½
1½	1½	117	150	111	139	103	127	97	121	92	115	87	110	82	104	74	92	70	86	65	81	1½	1½
1½	2	150	195	139	179	127	156	121	157	115	149	110	141	104	135	92	122	86	104	81	106	1½	2
2	2	195	278	179	254	156	237	157	226	149	208	141	196	135	191	122	179	104	167	106	156	2	2
2	2½	278	330	254	294	237	278	226	272	208	254	196	225	191	231	179	216	167	202	156	191	2	2½
2½	2½	330	404	294	381	278	364	272	341	254	324	225	312	231	295	216	271	202	254	191	242	2½	2½
2½	3	404	520	381	488	364	462	341	436	324	415	312	398	295	381	271	347	254	329	242	314	2½	3
3	3	520	728	488	676	462	630	436	600	415	577	398	548	381	532	347	497	329	462	314	445	3	3
3	3½	728	842	676	790	630	740	600	705	577	676	548	645	532	624	497	590	462	543	445	520	3	3½
3½	3½	842	1005	790	974	740	900	705	855	676	814	645	785	624	762	590	717	543	670	520	635	3½	3½
3½	4	1005	1150	974	1070	900	1020	855	970	814	925	785	888	762	854	717	808	670	775	635	718	3½	4
4	4	1150	1350	1070	1250	1020	1180	970	1120	925	1075	888	1030	854	981	808	924	775	877	718	832	4	4
4	5	.....	.....	1250	1580	1180	1490	1120	1420	1075	1375	1030	1320	981	1270	924	1190	877	1030	832	1070	4	5
5	5	.....	.....	1580	2150	1490	2050	1420	1960	1375	1901	1320	1850	1270	1780	1190	1690	1030	1600	1070	1500	5	5
5	6	.....	.....	2150	2600	2050	2510	1960	2400	1901	2260	1850	2280	1780	2200	1690	2080	1600	1965	1500	1780	5	6
6	6	.....	.....	2600	3280	2510	3240	2400	3120	2260	3060	2280	2960	2200	2880	2080	2700	1965	2540	1780	2310	6	6

Flow	Return	80	90	90	100	100	120	120	140	140	160	160	180	180	200	200	240	240	280	.....	.....	Flow	Return
¾	¾	0	13	0	11	0	9	0	9	0	8	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾	¾
¾	1	13	16	11	15	9	14	9	11	8	10	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	¾	1
1	1	16	23	15	22	14	20	11	17	10	16	0	15	0	14	0	13	0	11	.....	.....	1	1
1	1¼	23	32	22	30	20	26	17	24	16	23	15	21	14	20	13	18	11	17	.....	.....	1	1¼
1¼	1¼	32	51	30	47	26	44	24	39	23	37	21	35	20	33	18	33	17	31	.....	.....	1¼	1¼
1¼	1½	51	62	47	56	44	53	39	47	37	45	35	41	33	40	33	39	31	36	.....	.....	1¼	1½
1½	1½	62	75	56	69	53	67	47	60	45	58	41	52	40	52	39	46	36	40	.....	.....	1½	1½
1½	2	75	100	69	93	67	88	60	80	58	77	52	70	52	69	46	61	40	55	.....	.....	1½	2
2	2	100	150	93	144	88	127	80	121	77	115	70	110	69	104	61	92	55	86	.....	.....	2	2
2	2½	150	177	144	173	127	157	121	147	115	136	110	127	104	119	92	107	86	102	.....	.....	2	2½
2½	2½	177	214	173	219	157	208	147	185	136	167	127	150	119	138	107	127	102	121	.....	.....	2½	2½
2½	3	214	285	219	284	208	272	185	248	167	219	150	197	138	185	127	169	121	159	.....	.....	2½	3
3	3	285	428	284	404	272	392	248	370	219	312	197	288	185	277	169	248	159	231	.....	.....	3	3
3	3½	428	502	404	473	392	450	370	427	312	372	288	347	277	330	248	293	231	274	.....	.....	3	3½
3½	3½	502	607	473	578	450	532	427	497	372	463	347	427	330	404	293	358	274	335	.....	.....	3½	3½
3½	4	607	688	578	658	532	606	497	567	463	532	427	506	404	473	358	425	335	397	.....	.....	3½	4
4	4	688	796	658	762	606	705	567	660	532	624	506	590	473	575	425	520	397	485	.....	.....	4	4
4	5	796	1030	762	982	705	913	660	852	624	810	590	772	575	743	520	693	485	646	.....	.....	4	5
5	5	1030	1455	982	1387	913	1295	852	1210	810	1156	772	1120	743	1088	693	1030	646	981	.....	.....	5	5
5	6	1455	1790	1387	1720	1295	1620	1210	1520	1156	1450	1120	1420	1088	1340	1030	1290	981	1145	.....	.....	5	6
6	6	1790	2310	1720	2254	1620	2140	1520	2035	1450	1940	1420	1870	1340	1825	1290	1760	1145	1700	.....	.....	6	6

of the line must be known as well as the number of square feet of radiation that the section must carry. The length of a line is the distance, in feet, from the main heater risers to the farthest radiator risers, measured along the mains. In the case of the left-hand branch in Fig. 3, the length of the line is

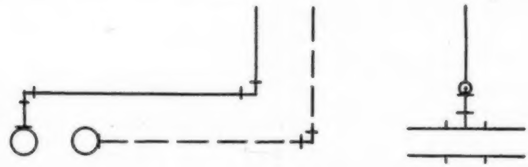

FIG. 4A. LATERAL SECTION OF RISERS (AT LEFT LOOKING ALONG THE MAINS, AT RIGHT LOOKING ACROSS THE MAINS)

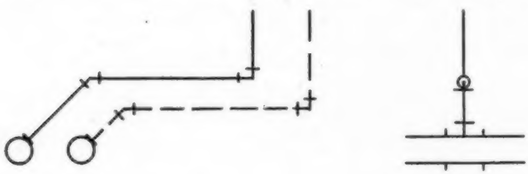

FIG. 4B. LATERAL SECTION OF RISERS (AT LEFT LOOKING ALONG THE MAINS, AT RIGHT LOOKING ACROSS THE MAINS)

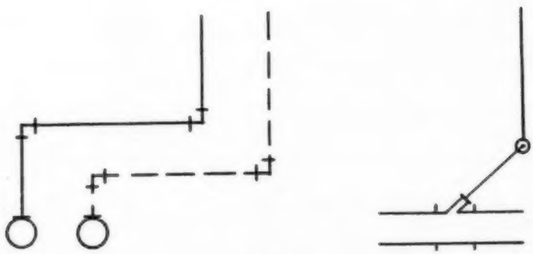

FIG. 4C. LATERAL SECTION OF RISERS (AT LEFT LOOKING ALONG THE MAINS, AT RIGHT LOOKING ACROSS THE MAINS)

8 ft + 23 ft + 23½ ft + 18 ft + 13 ft + 7 ft or 92½ ft. Thus, the length of the line lies between 90 ft and 100 ft.

Accordingly, to find the sizes of mains required between radiator No. 7 and radiator No. 6, first note that this section carries 57½ sq ft of radiation and then in the column headed "90-100" find the figures 49-60. The number of square feet required falls within this range and the correct sizes are found on the same line at the extreme left of the table. Thus, the correct size is 1½ in. for both the flow and return mains.

Proceeding in a similar manner, the remaining sizes are found to be as follows:

Section of Line Between Radiator	Required Capacity	Flow—In.	Return—In.
No. 7 and No. 6	57 sq ft	1½	1½
No. 6 and No. 5	114 sq ft	2	2
No. 5 and No. 4	147¼ sq ft	2	2½
No. 4 and No. 3	179¼ sq ft	2½	2½
No. 3 and No. 2	201¾ sq ft	2½	3
No. 2 and No. 1	264¾ sq ft	3	3
No. 1 and Heater	299¾ sq ft	3	3

Since the right-hand branch is 66½ ft long, its pipe sizes are found in the column headed "60-70." These sizes are as follows:

Section of Line Between Radiator	Required Capacity	Flow—In.	Return—In.
No. 10 and No. 9	75 sq ft	1½	2
No. 9 and No. 8	161¼ sq ft	2	2½
No. 8 and Heater	221¼ sq ft	2½	3

How to Select Sizes for the Main Risers

The main risers for the first branch must carry 299¾ sq ft of radiation. Therefore, by reference to Table 4 it will be noted that they should be at least 2½ in. The main risers for the right branch must carry 221¼ sq ft, and again, by reference to Table 4 it will be seen that they, too, should be at least 2½ in.

How to Select Orifice Sizes for Orifices to Be Made on the Job

In order to make allowance for the higher pressures that exist close to the heater, it is necessary to use smaller orifices near to the heater and larger ones farther away. To accomplish this, each line is divided into four sections of equal length. The section nearest the heater is No. 1, and the section farthest from the heater is No. 4. Thus, for the left hand branch in Fig. 3, each section is $\frac{92\frac{1}{2}}{4}$ or 23⅛ ft long.

Table 4. Capacities for Main Heater Risers[a]

Pipe Size—In.	Capacity in Sq Ft
1¼	0- 89
1½	89- 130
2	130- 220
2½	220- 325
3	325- 528
3½	528- 770
4	770-1000
5	1000-1660
6	1660-2480

[a]The pipe sizes for main heater risers may be larger than indicated in this table, but they must not be smaller.

TABLE 5—CAPACITIES OF ORIFICES IN SQUARE FEET OF DIRECT RADIATION

SECTION 1 — HEIGHT ABOVE MIDPOINT OF HEATER IN FEET

Diam. of Orifices in Inches	4½	5	5	6	6	7	7	8	8	9	9	10	10	11
3/32	....	....	....	....	....	....	....	....	....	....	....	....	....	....
7/64	6.6	8.6	7.3	9.4	7.8	10.2	8.4	10.9	8.9	11.6	9.4	12.3	9.9	12.9
1/8	8.6	10.8	9.4	11.8	10.2	12.8	10.9	13.8	11.6	14.6	12.3	15.4	12.9	16.1
9/64	10.8	13.4	11.8	14.7	12.8	15.8	13.8	17.0	14.6	18.0	15.4	19.1	16.1	20.0
5/32	13.4	16.1	14.7	17.7	15.8	19.2	17.0	20.4	18.0	21.6	19.1	22.8	20.0	24.2
11/64	16.1	18.7	17.7	20.7	19.2	21.8	20.4	24.5	21.6	26.0	22.8	27.5	24.2	28.8
3/16	18.7	22.0	20.7	24.8	21.8	26.8	24.5	28.8	26.0	30.6	27.5	32.4	28.8	33.8
13/64	22.0	26.0	24.8	28.6	26.8	31.0	28.8	33.2	30.6	35.0	32.4	37.2	33.8	39.0
7/32	26.0	30.0	28.6	33.1	31.0	35.4	33.2	37.2	35.0	40.1	37.2	43.0	39.0	45.0
15/64	30.0	33.6	33.1	37.0	35.4	40.0	37.2	43.0	40.1	46.0	43.0	48.7	45.0	51.1
1/4	33.6	42.9	37.0	47.1	40.0	51.7	43.0	55.5	46.0	59.0	48.7	62.4	51.1	65.3
9/32	42.9	52.8	47.1	58.1	51.7	63.2	55.5	67.6	59.0	71.9	62.4	76.0	65.3	80.0
5/16	52.8	63.0	58.1	69.6	63.2	75.7	67.6	81.5	71.9	85.5	76.0	92.0	80.0	96.5
11/32	63.0	75.1	69.6	81.8	75.7	91.0	81.5	97.5	85.5	104	92.0	110	96.5	115
3/8	75.1	86.1	81.8	98.0	91.0	107	97.5	114	104	122	110	128	115	134
13/32	86.1	103	98.0	114	107	124	114	131	122	140	128	148	134	155
7/16	103	117	114	130	124	139	131	149	140	159	148	168	155	172
15/32	117	132	130	146	139	159	149	171	159	182	168	193	172	203
1/2	132	166	146	185	159	201	171	216	182	230	193	264	....	....
9/16	166	205	185	228	201	248	....	....	....	....	....	....	....	....

Diam. of Orifices in Inches	11	12	12	13	13	14	14	15	15	16	16	17	17	18
3/32	....	....	7.5	10.6	7.8	11.1	8.1	11.5	8.4	11.9	8.7	12.2	8.9	12.6
7/64	10.4	13.5	10.6	13.8	11.1	14.3	11.5	14.9	11.9	15.4	12.2	15.9	12.6	16.4
1/8	13.5	16.8	13.8	17.4	14.3	18.1	14.9	18.7	15.4	19.4	15.9	20	16.4	20.8
9/64	16.8	21.0	17.4	21.6	18.1	22.5	18.7	23.3	19.4	24.2	20.	24.9	20.6	25.6
5/32	21.0	25.4	21.6	26.2	22.5	27.1	23.3	28.2	24.2	29.2	24.9	30.1	25.6	30.9
11/64	25.4	30.2	26.2	31.1	27.1	32.4	28.2	33.5	29.2	34.8	30.1	35.8	30.9	36.8
3/16	30.2	35.3	31.1	36.6	32.4	38.2	33.5	39.4	34.8	40.9	35.8	42.1	36.8	43.3
13/64	35.3	40.1	36.6	42.3	38.2	44.1	39.4	45.6	40.9	47.2	42.1	48.6	43.3	50.1
7/32	40.1	47.0	42.3	48.9	44.1	50.9	45.6	52.6	47.2	54.6	48.6	56.2	50.1	57.8
15/64	47.0	53.6	48.9	65.4	50.9	57.7	52.6	59.6	54.6	61.9	56.2	63.7	57.8	65.5
1/4	53.6	68.5	65.4	70.7	57.7	73.6	59.6	76.2	61.9	79	63.7	81.5	65.5	83.6
9/32	68.5	83.0	70.7	86.6	73.6	90.3	76.2	93.4	79	96.8	81.5	99.6	83.6	102
5/16	83.0	101	86.6	105	90.3	109	93.4	113	96.8	117	99.6	120	102	124
11/32	101	121	105	125	109	130	133	134	117	139	120	143	124	147
3/8	121	140	125	146	130	152	134	158	139	163	143	168	147	173
13/32	140	163	146	170	152	177	158	183	163	190	168	196	173	202
7/16	163	185	170	193	177	201	183	208	190	216	196	222	202	....
15/32	185	212	193	222	201	....	208	....	216	....	222	....	....	....
1/2	....	....	222	....	....	....	....	....	....	....	....	....	....	....
9/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....

Diam. of Orifices in Inches	18	20	20	22	22	24	24	26	26	28	28	30	30	33
3/32	9.4	13.3	9.9	1.4	10.3	14.5	10.7	15.2	11.1	15.6	11.5	16.2	11.8	17
7/64	13.3	17.3	1.4	18.2	14.5	19	15.2	19.8	15.6	20.3	16.2	21.1	17	22.1
1/8	17.3	21.8	18.2	22.8	19	23.8	19.8	24.8	20.3	25.5	21.1	26.5	22.1	27.8
9/64	21.8	26.1	22.8	28.4	23.8	29.7	24.3	30.9	25.5	31.8	26.5	33	27.8	34.6
5/32	26.1	32.7	28.4	34.4	29.7	35.9	30.9	37.4	31.8	38.4	33	39.8	34.6	41.8
11/64	32.7	39	34.4	40.9	35.9	42.7	37.4	44.5	38.4	45.6	39.8	46.5	41.8	49.7
3/16	39	45.8	40.9	48.2	42.7	50.2	44.5	52.4	45.6	53.8	46.5	55.9	49.7	58.5
13/64	45.8	53	48.2	55.6	50.2	58.1	52.4	60.5	53.8	62.1	55.9	64.5	58.5	67.6
7/32	53	61.2	55.6	64.3	58.1	67.1	60.5	69.9	62.1	71.8	64.5	74.5	67.6	78.1
15/64	61.2	69.4	64.3	73.9	67.1	76	69.9	79.2	71.8	81.4	74.5	84.5	78.1	88.6
1/4	69.4	88.5	73.9	93	76	97.1	79.2	101	81.4	104	84.5	108	88.6	113
9/32	88.5	108	93	114	97.1	119	101	124	104	127	108	132	113	139
5/16	108	131	114	138	119	144	124	150	127	154	132	160	139	167
11/32	131	156	138	164	144	171	150	178	154	183	160	190	167	199
3/8	156	183	164	192	171	201	178	209	183	215	190	223	199	234
13/32	183	213	192	224	201	....	209	....	215	....	223	....	234	....
7/16	213	....	224	....	....	....	....	....	....	....	....	....	....	....
15/32	....	....	....	....	....	....	....	....	....	....	....	....	....	....
1/2	....	....	....	....	....	....	....	....	....	....	....	....	....	....
9/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....

Diam. of Orifices in Inches	33	36	36	40	40	43	43	46	46	50	50	55	55	60
3/32	12.6	17.8	13.3	18.8	13.8	19.5	14.3	20.2	14.9	21.1	15.6	22.1	16.3	23.1
7/64	17.8	23.2	18.8	24.5	19.5	25.4	20.2	26.3	21.1	27.4	22.1	28.8	23.1	30
1/8	23.2	29.0	24.5	30.8	25.4	31.9	26.3	33	27.4	34.4	28.8	36.1	30	37.8
9/64	29.0	36.2	30.8	38.2	31.9	39.7	33	41.1	34.4	42.9	36.1	45	37.8	47
5/32	36.2	43.7	38.2	46.2	39.7	48	41.1	49.6	42.9	51.8	45	54.4	47	56.9
11/64	43.7	52	46.2	55	48	57.2	49.6	59.1	51.8	61.7	54.4	64.6	56.9	67.6
3/16	52	61.2	55	64.8	57.2	67.2	59.1	69.5	61.7	72.6	64.6	76.1	67.6	79.6
13/64	61.2	70.8	64.8	74.9	67.2	77.6	69.5	80.4	72.6	83.9	76.1	88	79.6	92
7/32	70.8	81.8	74.9	86.5	77.6	89.8	80.4	92.9	83.9	96.9	88	102	92	106
15/64	81.8	92.7	86.5	98	89.8	102	92.9	105	96.9	110	102	115	106	120
1/4	92.7	118	98	125	102	130	105	134	110	140	115	147	120	154
9/32	118	145	125	153	130	159	134	165	140	172	147	180	154	188
5/16	145	175	153	185	159	192	165	199	172	207	180	218	188	227
11/32	175	208	185	220	192	229	199	229	207	....	218	....	227	....
3/8	208	....	220	....	229	....	229	....	....	....	....	....	....	....
13/32	....	....	....	....	....	....	....	....	....	....	....	....	....	....
7/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....
15/32	....	....	....	....	....	....	....	....	....	....	....	....	....	....
1/2	....	....	....	....	....	....	....	....	....	....	....	....	....	....
9/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....

TABLE 5. CAPACITIES OF ORIFICES IN SQUARE FEET OF DIRECT RADIATION—*Continued*

SECTION 2 — HEIGHT ABOVE MIDPOINT OF HEATER IN FEET — SECTION 2

Diam. of Orifices in Inches	4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18
Two																												
3/32	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	5.9	8.4	6.2	8.7	6.4	9.1	6.6	9.4	6.9	9.7	7.1	10.1
7/64	5.1	6.6	5.6	7.3	6.1	7.9	6.6	8.6	7.0	9.1	7.4	9.7	7.8	10.2	8.2	10.7	8.4	10.9	8.7	11.4	9.1	11.8	9.4	12.2	9.7	12.6	10.1	13.
1/8	6.6	8.2	7.3	9.1	7.9	10.	8.6	10.8	9.1	11.5	9.7	12.1	10.2	12.8	10.7	13.4	10.9	13.7	11.4	14.3	11.8	14.8	12.2	15.3	12.6	15.8	13.2	16.3
9/64	8.2	10.2	9.1	11.4	10.	12.4	10.8	13.5	11.5	14.2	12.1	15.1	12.8	15.7	13.4	16.5	13.7	17.1	14.3	17.8	14.8	18.5	15.3	19.1	15.8	19.7	16.3	20.4
5/32	10.2	12.4	11.4	13.8	12.4	15.	13.5	16.	14.2	17.1	15.1	18.1	15.7	19.1	16.5	20.	17.1	20.6	17.8	21.5	18.5	22.3	19.1	23.1	19.7	23.8	20.4	24.6
11/64	12.4	14.7	13.8	16.1	15.	17.6	16.	19.	17.1	20.4	18.1	21.2	19.1	22.4	20.0	23.8	20.6	24.6	21.5	25.6	22.3	26.6	23.1	27.5	23.8	28.4	24.6	29.3
3/16	14.7	17.	16.1	19	17.6	20.7	19.0	22.0	20.4	23.5	21.2	25.4	22.4	26.6	23.8	28.	24.6	28.9	25.6	30.1	26.6	31.2	27.5	32.3	28.4	33.4	29.3	34.4
13/64	17.	19.7	19.	21.4	20.7	23.5	22.	25.9	23.5	27.8	25.4	29.4	26.6	30.9	28.	32.4	28.9	33.4	30.1	34.8	31.2	36.1	32.3	37.3	33.4	38.6	34.4	39.8
7/32	19.7	22.1	21.4	25.4	23.5	27.7	25.9	29.9	27.8	32.0	29.4	33.6	30.9	35.4	32.4	37.2	33.4	39.3	34.8	40.1	36.1	41.7	37.3	43.1	38.6	44.6	39.8	46.
15/64	22.1	25.7	25.4	28.7	27.7	31.4	29.9	33.4	32.0	35.7	33.6	38.	35.4	40.	37.2	42.	39.3	43.8	40.1	45.6	41.7	47.3	43.1	48.9	44.6	50.6	46.	52.1
1/4	25.7	32.8	28.7	36.	31.4	39.4	33.4	42.7	35.7	45.1	38.	48.6	40.	51.4	42.	54.	43.8	55.2	45.6	57.5	47.3	60.4	48.9	62.5	50.6	64.5	52.1	66.6
9/32	32.8	39.3	36.	44.	39.4	48.7	42.7	52.5	45.1	56.1	48.6	59.5	51.4	62.9	54.	65.5	55.2	68.5	57.5	71.3	60.4	74.	62.5	76.5	64.5	79.	66.6	81.6
5/16	39.3	46.2	44.	53.5	48.7	58.9	52.5	63.4	56.1	67.4	59.5	71.5	62.9	75.6	65.5	79.2	68.5	82.6	71.3	86.	74.	89.4	76.5	92.5	79.	95.5	81.6	99.4
11/32	46.2	57.1	53.5	63.2	58.9	69.2	63.4	75.0	67.4	81.	71.5	83.5	75.6	90.	79.2	95.	82.6	98.4	86.	102.	89.4	106.	92.5	110.	95.5	114.	99.4	117.
3/8	57.1	66.	63.2	74.1	69.2	81.3	75.0	86.	81.	95.	83.5	100.	90.	106.	95.	111.	98.4	115.	102.	120.	106.	125.	110.	124.	114.	134.	117.	138.
13/32	66	77.	74.1	83.6	81.3	95.6	86.	103.	95.	110.	100.	117.	106.	123.	111.	130.	115.	135.	120.	140.	125.	145.	124.	150.	134.	155.	138.	160.
7/16	77	84.	83.6	99.0	95.6	108.	103.	117.	110.	125	117.	131.	123.	138.	130.	145.	135.	153.	140.	159.	145.	165.	150.	171.	155.	177.	160.	182.
15/32	84	102.	99.	114.	108.	125.	117.	131.	125.	141.	131.	150.	138.	159.	145.	166.	153.	....	159.	....	165.	....	171.	....	177.	....	182.	....
1/2	102	128.	114.	138.	125.	152.	131.	166.	141.	178.	150.	190.	159.	200.	166.	210.	....	....	....	....	....	....	....	....	....	....	....	....
9/16	128	151.	138.	172.	152.	189.	166.	204.	178.	220.	190.	234.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
5/8	151	184.	172.	208.	189.	228.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
11/16	184	219.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
3/4	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....

Diam. of Orifices in Inches	18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	33	40	40	43	43	46	46	50	50	55	55	60
Two																												
3/32	7.5	10.6	7.9	11.1	8.2	11.6	8.6	12.2	8.9	12.7	9.3	13.1	9.7	13.8	10.2	14.4	10.7	15.2	11.1	15.8	11.5	16.3	12.	17.	12.6	17.9	13.2	18.7
7/64	10.6	13.8	11.1	14.4	11.6	15.1	12.2	15.8	12.7	16.5	13.1	17.1	13.8	17.9	14.4	18.7	15.2	19.7	15.8	20.5	16.3	21.2	17.	22.2	17.9	23.2	18.7	24.3
1/8	13.8	17.3	14.4	18.2	15.1	19.	15.8	19.9	16.5	20.7	17.1	21.4	17.9	22.4	18.7	23.5	19.7	24.8	20.5	25.8	21.2	27.6	22.2	27.8	23.2	29.2	24.3	30.5
9/64	17.3	21.6	18.2	22.6	19.0	23.6	19.9	24.8	20.7	25.8	21.4	26.6	22.4	28.	23.5	29.2	24.8	30.8	25.8	32.1	27.6	33.2	27.8	34.6	29.2	36.4	30.5	38.
5/32	21.6	26.1	22.6	27.3	23.6	28.6	24.8	30.	25.8	31.2	26.6	32.2	28.	33.8	29.2	35.4	30.8	37.3	32.1	38.8	33.2	40.1	34.6	41.9	36.4	44.	38.	46.
11/64	26.1	31.	27.3	32.5	28.6	34.	30.	35.6	31.2	37.1	32.2	38.4	33.8	40.2	35.4	42.1	37.3	44.4	38.8	46.1	40.1	47.7	41.9	49.9	44.	52.4	46.	54.6
3/16	31.	36.5	32.5	38.2	34.	40.	35.6	42.	37.1	43.6	38.4	45.1	40.2	47.4	42.1	49.5	44.4	52.5	46.1	54.4	47.7	56.1	49.9	58.7	52.4	61.5	54.6	64.4
13/64	36.5	42.2	38.2	44.2	40.	46.3	42.	48.5	43.6	50.4	45.1	52.1	47.4	54.6	49.5	57.2	52.2	60.4	54.4	62.6	56.1	65.	58.7	67.8	61.5	71.1	64.4	74.4
7/32	42.2	48.7	44.2	51.1	46.3	53.5	48.5	56.	50.4	58.2	52.1	60.2	54.6	63.2	57.2	66.1	60.4	69.6	62.6	72.5	65.	75.	67.8	78.4	71.1	82.1	74.4	86.
15/64	48.7	55.3	51.1	57.9	53.5	60.6	56.	63.5	58.2	66.	60.2	68.3	63.2	71.5	66.1	75.	69.6	79.	72.5	82.1	75.	85.	78.4	88.8	82.1	93.1	86.	97.5
1/4	55.3	70.5	57.9	74.	60.6	77.4	63.5	81.	66.	84.3	68.3	87.1	71.5	91.5	75.	95.8	79.	101.	82.1	105.	85.	109.	88.8	113.	93.1	119.	97.5	124.
9/32	70.5	86.5	74.	90.6	77.4	94.8	81.	99.4	84.3	103.	87.1	107.	91.5	112.	95.8	117.	101.	124.	105.	129.	109.	133.	113.	139.	119.	146.	124.	152.
5/16	86.5	104.	90.6	109.	94.8	114.	99.4	120.	103.	125.	107.	129.	112.	135.	117.	142.	124.	149.	129.	155.	133.	160.	139.	168.	146.	176.	152.	184.
11/32	104.	124.	109.	130.	114.	136.	120.	143.	125.	148.	129.	154.	135.	161.	142.	169.	149.	178.	155.	185.	160.	191.	168.	200.	176.	210.	184.	219.
3/8	124.	146.	130.	153.	136.	160.	143.	168.	148.	174.	154.	180.	161.	189.	169.	198.	178.	208.	185.	217.	191.	224.	200.	....	210.	....	219.	....
13/32	146.	170.	153.	178.	160.	186.	168.	195.	174.	203.	180.	210.	189.	220.	198.	230.	208.	....	217.	....	224.	....	....	....	....	....	....	....
7/16	170.	193.	178.	202.	186.	212.	195.	222.	203.	....	210.	....	220.	....	230.	....	....	....	....	....	....	....	....	....	....	....	....	....
15/32	193.	....	202.	....	212.	....	222.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
1/2	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
9/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
5/8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
11/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....
3/4	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....

TABLE 5. CAPACITIES OF ORIFICES IN SQUARE FEET OF DIRECT RADIATION—*Continued*

Diam. of Orifices in Inches	Section 3														Height Above Midpoint of Heaters in Feet													Section 3	Diam. of Orifices in Inches
	4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18	
3/32	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	3/32
7/64	3.6	4.8	4.1	5.3	4.5	5.9	4.9	6.4	5.3	6.8	5.6	7.3	5.9	7.8	6.2	8.1	6.3	8.2	6.6	8.5	6.8	8.9	7.2	9.2	7.4	9.6	7.6	9.9	7/64
1/8	4.8	6.0	5.3	6.7	5.9	7.4	6.4	8.0	6.8	8.6	7.3	9.2	7.8	9.7	8.1	10.1	8.2	10.3	8.5	10.7	8.9	11.2	9.2	11.6	9.6	12.0	9.9	12.4	1/8
9/64	6.0	7.4	6.7	8.4	7.4	9.2	8.0	10.0	8.6	10.7	9.2	11.4	9.7	12.0	10.1	12.6	10.3	12.8	10.7	13.4	11.2	13.9	11.6	14.4	12.0	15.0	12.4	15.4	9/64
5/32	7.4	9.0	8.4	10.1	9.2	11.1	10.0	12.0	10.7	12.9	11.4	13.7	12.0	14.5	12.6	15.2	12.8	15.5	13.4	16.2	13.9	16.8	14.4	17.5	15.0	18.1	15.4	18.6	5/32
11/64	9.0	10.7	10.1	12.0	11.1	13.2	12.0	14.4	12.9	15.4	13.7	16.1	14.5	17.1	15.2	18.0	15.5	18.4	16.2	19.2	16.8	20.0	17.5	20.8	18.1	21.5	18.6	22.2	11/64
3/16	10.2	12.6	12.0	14.1	13.2	15.6	14.4	16.6	15.4	17.8	16.1	19.0	17.1	20.1	18.0	21.1	18.4	21.7	19.2	22.6	20.0	23.6	20.8	24.5	21.5	25.3	22.2	26.2	3/16
13/64	12.6	14.5	14.1	15.9	15.6	17.5	16.6	19.1	17.8	20.6	19.0	21.4	20.1	22.6	21.1	24.4	21.7	25.1	22.6	26.2	23.6	27.2	24.5	28.2	25.3	29.2	26.2	30.2	13/64
7/32	14.5	16.1	15.9	18.3	17.5	20.3	19.1	21.4	20.6	23.1	21.4	25.4	22.6	26.8	24.4	28.2	25.1	28.9	26.2	30.2	27.2	31.4	28.2	32.6	29.2	33.8	30.2	34.8	7/32
15/64	16.1	18.3	18.3	20.8	20.3	22.2	21.4	25.1	23.1	27.0	25.4	28.8	26.8	30.4	28.2	32.0	28.9	32.8	30.2	34.2	31.4	35.7	32.6	37.0	33.8	38.3	34.8	39.5	15/64
1/4	18.3	22.0	20.8	26.6	22.2	29.4	25.1	32.0	27.0	33.1	28.8	36.0	30.4	38.2	32.0	40.2	32.8	41.8	34.2	43.7	35.7	45.5	37.0	47.2	38.3	48.9	39.5	50.5	1/4
9/32	22.0	29.0	26.6	32.9	29.4	35.4	32.0	38.6	33.1	41.8	36.0	43.9	38.2	46.6	40.2	50.3	41.8	51.3	43.7	53.6	45.5	55.7	47.2	57.9	48.9	59.9	50.5	61.9	9/32
5/16	29.0	33.2	32.9	38.2	35.4	42.6	38.6	42.2	41.8	50.7	43.9	54.1	46.6	57.4	50.3	60.4	51.3	61.9	53.6	64.6	55.7	67.3	57.9	70.0	59.9	72.4	61.9	74.6	5/16
11/32	33.2	37.6	38.2	43.6	42.6	51.0	42.2	53.6	50.7	60.0	54.1	63.2	57.4	67.0	60.4	70.5	61.9	73.7	64.6	77.0	67.3	80.2	70.0	83.2	72.4	86.1	74.6	88.9	11/32
3/8	37.6	47.3	43.6	54.4	51.0	60.0	55.6	64.4	60.0	69.5	63.2	74.2	67.0	78.7	70.5	81.0	73.7	86.5	77.0	90.4	80.2	94.0	83.2	97.6	86.1	101.	88.9	104.	3/8
13/32	47.3	54.6	54.4	62.4	60.0	68.1	64.4	74.4	69.5	80.4	74.2	83.4	78.7	91.8	81.0	97.4	86.5	101.	90.4	105.	94.0	110.	97.6	114.	101.	118.	104.	122.	13/32
7/16	54.6	62.4	62.4	69.6	68.1	77.6	74.4	81.5	80.4	92.7	83.4	99.0	91.8	105.	97.4	110.	101.	114.	105.	119.	110.	124.	114.	129.	118.	134.	122.	138.	7/16
15/32	62.4	69.5	69.6	80.0	77.6	84.8	81.5	93.8	92.7	106.	99.0	116.	105.	120.	110.	126.	114.	131.	119.	137.	124.	142.	129.	148.	134.	153.	138.	158.	15/32
1/2	69.5	80.6	80.0	104.	84.8	114.	93.8	125.	106.	129.	116.	138.	120.	144.	126.	155.	131.	165.	137.	173.	142.	180.	148.	187.	153.	194.	158.	200.	1/2
9/16	80.6	111.	104.	128.	114.	134.	125.	147.	129.	160.	138.	172.	144.	183.	155.	192.	165.	205.	173.	214.	180.	222.	187.	231.	194.	239.	200.	....	9/16
5/8	111.	134.	128.	193.	134.	162.	147.	178.	160.	195.	172.	207.	183.	226.	192.	232.	205.	....	214.	....	222.	....	231.	....	239.		....	....	5/8
11/16	134.	145.	193.	172.	162.	194.	178.	211.	195.	233.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11/16
3/4	145.	148.	172.	174.	194.	196.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	3/4
7/8	148.	201.	174.	232.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	7/8

	18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60	
3/32	....	....	....	....	....	....	....	....	6.8	9.7	7.1	10.0	7.5	10.6	7.8	11.0	8.3	11.6	8.6	12.1	8.9	12.6	9.3	13.2	9.8	13.8	10.2	14.5	3/32
7/64	8.0	10.5	8.5	11.0	8.9	11.6	9.3	12.1	9.7	12.6	10.0	13.0	10.6	13.7	11.0	14.4	11.6	15.2	12.1	15.8	12.6	16.4	13.2	17.1	13.8	18.0	14.5	18.8	7/64
1/8	10.5	13.1	11.0	13.9	11.6	14.5	12.1	15.2	12.6	15.8	13.0	16.4	13.7	17.2	14.4	18.0	15.2	19.1	15.8	19.8	16.4	20.6	17.1	21.5	18.0	22.6	18.8	23.6	1/8
9/64	13.1	16.4	13.9	17.3	14.5	18.1	15.2	18.9	15.8	19.6	16.4	20.4	17.2	21.5	18.0	22.5	19.1	23.8	19.8	24.7	20.6	25.6	21.5	26.7	22.6	28.1	23.6	29.4	9/64
5/32	16.4	19.8	17.3	20.9	18.1	21.9	18.9	22.8	19.6	23.8	20.4	24.6	21.5	26.0	22.5	27.2	23.8	28.8	24.7	29.9	25.6	31.0	26.7	32.3	28.1	33.9	29.4	35.5	5/32
11/64	19.8	23.5	20.9	24.8	21.9	26.0	22.8	27.1	23.8	28.2	24.6	29.3	26.0	30.8	27.2	32.3	28.8	34.2	29.9	35.5	31.0	35.8	32.3	38.4	33.9	40.4	35.5	42.3	11/64
3/16	23.5	27.7	24.8	29.2	26.0	30.6	27.1	32.0	28.2	33.3	29.3	34.5	30.8	36.4	32.3	38.0	34.2	40.2	35.5	41.8	35.8	43.4	38.4	45.2	40.4	47.5	42.3	49.7	3/16
13/64	27.7	32.0	29.2	33.8	30.6	35.4	32.0	37.0	33.3	38.5	34.5	39.9	36.4	42.0	38.0	44.0	40.2	46.5	41.8	48.4	43.4	50.1	45.2	52.3	47.5	54.9	49.7	57.5	13/64
7/32	32.0	36.9	33.8	39.0	35.4	40.8	37.0	42.6	38.5	44.4	39.9	46.1	42.0	48.5	44.0	50.7	46.5	53.7	48.4	55.8	50.1	57.8	52.3	60.4	54.9	63.4	57.5	66.4	7/32
15/64	36.9	41.9	39.0	44.2	40.8	46.4	42.6	48.4	44.4	50.4	46.1	52.2	48.5	55.0	50.7	57.6	53.7	61.0	55.8	63.2	57.8	65.6	60.4	68.5	63.4	72.0	66.4	75.3	15/64
1/4	41.9	53.5	44.2	56.5	46.4	59.2	48.4	61.8	50.4	64.3	52.2	66.7	55.0	70.2	57.6	73.5	61.0	77.8	63.2	80.7	65.6	83.7	68.5	87.5	72.0	91.9	75.3	96.1	1/4
9/32	53.5	65.5	56.5	69.1	59.2	72.5	61.8	75.7	64.3	78.8	66.7	81.7	70.2	86.0	73.5	90.0	77.8	95.4	80.7	99.0	83.7	103.	87.5	108.	91.9	113.	96.1	118.	9/32
5/16	65.5	79.1	69.1	83.5	72.5	87.5	75.7	91.4	78.8	95.0	81.7	98.6	86.0	104.	90.0	109.	95.4	115.	99.0	119.	103.	124.	108.	129.	113.	136.	118.	142.	5/16
11/32	79.1	94.2	83.5	99.4	87.5	104.	91.4	109.	95.0	113.	98.6	116.	104.	124.	109.	129.	115.	137.	119.	142.	124.	147.	129.	154.	136.	162.	142.	169.	11/32
3/8	94.2	111.	99.4	117.	104.	122.	109.	128.	113.	133.	116.	138.	124.	145.	129.	152.	137.	161.	142.	167.	147.	173.	154.	181.	162.	190.	169.	199.	3/8
13/32	111.	129.	117.0	136.	122.	142.	128.	149.	133.	155.	138.	160.	145.	169.	152.	177.	161.	187.	167.	194.	173.	202.	181.	210.	190.	221.	199.	231	13/32
7/16	129.	146.	136.	154.	142.	162.	149.	169.	155.	176.	160.	182.	169.	192.	177.	201.	187.	212.	194.	220.	202.	....	210.	....	221.	....	231.	....	7/16
15/32	146.	168.	154.	177.	162.	185.	169.	194.	176.	201.	182.	209.	192.	220.	201.	....	212.	....	220.	....	....	....	....	....	....	....	....	....	15/32
1/2	168.	212.	177.	224.	185.	234.	194.	245.	201.	....	209.	....	220.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1/2
9/16	212.	....	224.	....	234.	....	245.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9/16
5/8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	5/8
11/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11/16
3/4	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	3/4
7/8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	7/8

TABLE 5. CAPACITIES OF ORIFICES IN SQUARE FEET OF DIRECT RADIATION—*Continued*

Diam. of Orifices in Inches	Section 4													Height Above Midpoint of Heater in Feet														Section 4	Diam. of Orifices in Inches
	4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18	
7/64	2.5	3.3	2.9	3.7	3.2	4.2	3.5	4.5	3.7	4.9	4.0	5.2	4.2	5.5	4.5	5.8	4.5	5.9	4.6	6.0	4.7	6.1	4.9	6.4	5.1	6.7	5.3	6.9	7/64
1/8	3.3	4.1	3.7	4.7	4.2	5.2	4.5	5.7	4.9	6.1	5.2	6.5	5.5	6.9	5.8	7.3	5.9	7.3	6.0	7.4	6.1	7.7	6.4	8.0	6.7	8.4	6.9	8.7	1/8
9/64	4.1	5.1	4.7	5.9	5.2	6.5	5.7	7.1	6.1	7.7	6.5	8.2	6.9	8.9	7.3	9.1	7.3	9.2	7.4	9.3	7.7	9.7	8.0	10.0	8.4	10.4	8.7	10.8	9/64
5/32	5.1	6.2	5.9	7.1	6.5	7.9	7.1	8.6	7.7	9.2	8.2	9.8	8.9	10.4	9.1	11.0	9.2	11.0	9.3	11.1	9.7	11.6	10.0	12.1	10.4	12.6	10.8	13.0	5/32
11/64	6.2	7.4	7.1	8.4	7.9	9.4	8.6	10.2	9.2	11.0	9.8	11.7	10.4	12.4	11.0	13.1	11.0	13.2	11.1	13.2	11.6	13.8	12.1	14.4	12.6	15.0	13.0	15.5	11/64
3/16	7.4	8.7	8.4	9.9	9.4	11.0	10.2	12.0	11.0	12.9	11.7	13.8	12.4	14.6	13.1	15.4	13.2	15.6	13.2	15.5	13.8	16.3	14.4	16.9	15.0	17.6	15.5	18.2	3/16
13/64	8.7	10.0	9.9	11.4	11.0	12.7	12.0	13.9	12.9	14.9	13.8	15.2	14.6	16.2	15.4	17.1	15.6	17.3	15.5	17.9	16.3	18.8	16.9	19.6	17.6	20.4	18.2	21.1	13/64
7/32	10.0	11.5	11.4	13.2	12.7	14.7	13.9	15.1	14.9	16.4	15.2	17.6	16.2	18.7	17.1	19.8	17.3	20.0	17.9	20.7	18.8	21.7	19.6	22.6	20.4	23.5	21.1	24.4	7/32
15/64	11.5	13.1	13.2	15.0	14.7	15.5	15.1	17.1	16.4	18.6	17.6	20.0	18.7	21.2	19.8	21.8	20.0	22.4	20.7	23.5	21.7	24.6	22.6	25.6	23.5	26.7	24.4	27.6	15/64
1/4	13.1	14.8	15.0	18.2	15.5	19.8	17.1	20.4	18.6	22.4	20.0	25.4	21.2	27.2	21.8	28.7	22.4	28.5	23.5	30.0	24.6	31.4	25.6	32.7	26.7	34.0	27.6	35.2	1/4
9/32	14.8	18.2	18.2	21.4	19.8	22.2	20.4	26.8	22.4	29.0	25.4	31.2	27.2	33.3	28.7	34.0	28.5	35.5	30.0	36.8	31.4	38.4	32.7	40.1	34.0	41.8	35.2	43.2	9/32
5/16	18.2	22.0	21.4	23.2	22.2	29.3	26.8	32.4	29.0	33.6	31.2	36.2	33.2	38.8	34.0	41.1	35.0	42.3	36.8	44.4	38.4	46.5	40.1	48.4	41.8	50.5	43.2	52.1	5/16
11/32	22.0	26.2	23.2	30.8	29.3	32.6	32.4	36.5	33.6	40.0	36.2	43.1	38.8	44.7	41.1	49.6	42.3	50.3	44.4	52.8	46.5	55.3	48.4	57.7	50.5	60.1	52.1	62.0	11/32
3/8	26.2	30.7	30.8	33.2	32.6	38.7	36.5	42.8	40.0	45.0	43.1	51.1	44.7	56.1	49.6	57.3	50.3	59.1	52.8	62.0	55.3	65.0	57.7	67.7	60.1	70.5	62.0	72.9	3/8
13/32	30.7	31.6	33.2	38.7	38.7	44.6	42.8	51.3	45.0	55.7	51.7	60.2	55.1	62.9	57.3	66.6	59.1	68.8	62.0	72.2	65.0	75.6	67.7	78.8	70.5	82.0	72.9	84.8	13/32
7/16	31.6	35.8	38.7	43.8	44.6	52.4	51.3	58.1	55.7	63.4	60.2	66.6	62.9	71.4	66.6	75.6	68.8	78.0	72.2	82.0	75.6	85.8	78.8	89.5	82.0	93.2	84.8	96.4	7/16
15/32	35.8	41.2	43.8	46.1	52.4	60.1	58.1	64.5	63.4	70.8	66.6	76.4	71.4	81.9	75.6	82.3	78.0	89.5	82.0	94.0	85.8	98.2	89.5	102.0	93.2	106.0	96.4	111.0	15/32
1/2	41.2	45.2	46.1	64.0	60.1	72.8	64.5	81.5	70.8	91.8	76.4	98.5	81.9	105.0	82.3	111.0	89.5	113.0	94.0	119.0	98.2	124.0	102.0	129.0	106.0	135.0	111.0	141.0	1/2
9/16	45.2	63.2	64.0	77.6	72.8	79.0	81.5	104.0	91.8	113.0	98.5	123.0	105.0	130.0	111.0	130.0	113.0	139.0	119.0	147.0	124.0	153.0	129.0	160.0	135.0	166.0	141.0	172.0	9/16
5/8	63.2	76.2	77.6	95.2	79.0	112.0	104.0	125.0	113.0	136.0	123.0	137.0	130.0	146.0	130.0	157.0	139.0	168.0	147.0	177.0	153.0	185.0	160.0	193.0	166.0	201.0	172.0	208.0	5/8
11/16	76.2	97.2	95.2	117.0	112.0	134.0	125.0	133.0	136.0	148.0	137.0	163.0	146.0	176.0	157.0	187.0	168.0	201.0	177.0	211.5	185.0	221.0	193.0	230.0	....	....	....	....	11/16
3/4	97.2	132.0	117.0	131.0	134.0	158.0	133.0	184.0	148.0	201.0	163.0	220.0	176.0	238.0	187.0	254.0	....	....	....	....	....	....	....	....	....	....	....	....	3/4
7/8	132.0	152.0	131.0	191.0	158.0	210.0	184.0	240.0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	7/8
1	152.0	191.0	191.0	252.0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1

	18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60	
7/64	5.7	7.3	6.0	7.8	6.3	8.2	6.6	8.6	6.9	9.0	7.2	9.3	7.6	9.9	7.9	10.3	8.4	11.0	8.8	11.4	9.1	11.8	9.5	12.6	10.0	13.0	10.5	13.6	7/64
1/8	7.3	9.3	7.8	9.8	8.2	10.3	8.6	10.8	9.0	11.3	9.3	11.7	9.9	12.4	10.3	13.0	11.0	13.8	11.4	14.3	11.8	14.8	12.6	15.5	13.0	16.3	13.6	17.1	1/8
9/64	9.3	11.5	9.8	12.2	10.3	12.8	10.8	13.4	11.3	14.1	11.7	14.6	12.4	15.4	13.0	16.1	13.8	17.1	14.3	17.8	14.8	18.5	15.5	19.3	16.3	20.3	17.1	21.2	9/64
5/32	11.5	13.9	12.2	14.8	12.8	15.5	13.4	16.2	14.1	17.0	14.6	17.6	15.4	18.6	16.1	19.5	17.1	20.7	17.8	21.5	18.5	22.3	19.3	23.4	20.3	24.5	21.2	25.7	5/32
11/64	13.9	16.5	14.8	17.6	15.5	18.5	16.2	19.3	17.0	20.2	17.6	20.9	18.6	22.2	19.5	23.2	20.7	24.6	21.5	26.6	22.3	26.6	23.4	27.8	24.5	29.2	25.7	30.6	11/64
3/16	16.5	19.4	17.6	20.6	18.5	21.7	19.3	22.7	20.2	23.8	20.9	24.7	22.2	26.1	23.2	27.3	24.6	29.0	25.6	30.2	26.6	31.2	27.8	32.7	29.2	33.4	30.6	36.0	3/16
13/64	19.4	22.5	20.6	23.8	21.7	25.1	22.7	26.2	23.8	27.5	24.7	28.5	26.1	30.1	27.3	31.5	29.0	33.5	30.2	34.8	31.2	36.1	32.7	37.8	33.4	39.4	36.0	41.5	13/64
7/32	22.5	26.0	23.8	27.5	25.1	29.0	26.2	30.3	27.5	31.7	28.5	32.9	30.1	34.8	31.5	36.4	33.5	38.7	34.8	40.2	36.1	41.7	37.8	43.7	39.7	45.9	41.5	48.0	7/32
15/64	26.0	26.0	27.5	31.2	29.0	32.9	30.3	34.4	31.7	36.0	32.9	37.3	34.8	39.4	36.4	41.3	38.7	43.9	40.2	45.6	41.7	47.3	43.7	49.5	45.9	52.0	48.0	54.4	15/64
1/4	29.4	29.4	31.2	39.8	32.9	42.0	34.4	43.4	36.0	45.9	37.3	47.6	39.4	50.4	41.3	52.6	43.9	56.0	45.6	58.2	47.3	60.4	49.5	63.1	52.0	66.4	54.4	69.5	1/4
9/32	37.5	37.5	39.8	48.9	42.0	51.4	43.4	53.8	45.9	56.3	47.6	58.4	50.4	61.7	52.6	64.6	56.0	68.7	58.2	71.4	60.4	74.0	63.1	77.4	66.4	81.4	69.5	86.1	9/32
5/16	46.0	46.0	48.9	59.0	51.4	62.0	53.8	64.2	56.3	68.0	58.4	70.5	61.7	74.5	64.6	77.9	68.7	83.0	71.4	86.1	74.0	89.3	77.4	93.5	81.4	98.3	85.1	103.0	5/16
11/32	55.5	55.5	59.0	70.3	62.0	74.0	64.2	79.5	68.0	70.9	70.5	84.1	74.5	88.1	77.9	92.8	83.0	98.9	86.1	103.0	89.3	106.0	93.5	111.0	98.3	117.0	103.0	123.0	11/32
3/8	66.1	66.1	70.3	82.5	74.0	87.0	79.5	90.8	70.9	95.0	84.1	98.6	88.1	104.0	92.8	109.0	98.9	116.0	103.0	120.0	106.0	125.0	111.0	131.0	117.0	137.0	123.0	144.0	3/8
13/32	77.8	77.8	82.5	96.0	87.0	101.0	90.8	106.0	95.0	111.0	98.6	115.0	104.0	121.0	109.0	127.0	116.0	135.0	120.0	140.0	125.0	145.0	131.0	152.0	137.0	160.0	144.0	167.0	13/32
7/16	90.5	90.5	96.0	109.0	101.0	115.0	106.0	120.0	111.0	125.0	115.0	130.0	121.0	138.0	127.0	144.0	135.0	154.0	140.0	159.0	145.0	165.0	158.0	173.0	160.0	181.0	167.0	190.0	7/16
15/32	103.0	118.0	109.0	125.0	115.0	132.0	120.0	138.0	125.0	144.0	130.0	149.0	138.0	158.0	144.0	165.0	154.0	175.0	159.0	182.0	165.0	190.0	173.0	198.0	181.0	208.0	190.0	218.0	15/32
1/2	118.0	149.0	125.0	157.0	132.0	166.0	138.0	174.0	144.0	182.0	149.0	189.0	158.0	200.0	165.0	204.0	175.0	220.0	182.0	230.0	190.0	240.0	198.0	250.0	....	....	....	....	1/2
9/16	149.0	184.0	157.0	195.0	166.0	205.0	174.0	215.0	182.0	224.0	189.0	233.0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9/16
5/8	184.0	222.0	195.0	235.0	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	5/8
11/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11/16
3/4	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	3/4
7/8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	7/8
1	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1

This distributes the radiators among the sections as follows:

Section No. 1 contains radiator No. 1
Section No. 2 contains radiators No. 2 and No. 3
Section No. 3 contains radiators No. 4 and No. 5
Section No. 4 contains radiators No. 6 and No. 7

In addition to the section in which a radiator is located, it is necessary to know its height above the mid-point of the heater. Since it has been decided to use the high flow connection the distance above the mid-point of the heater must be measured to the mid-point of the radiator. In the system under consideration the radiators are all 26 in. and their mid-points are all 7 ft 2 in. above the mid-point of the heater. Consequently, in Table 5 only the column headed "7-8" is of interest in this case. The actual selections can now be made.

Radiator No. 1 has a surface of 35 sq ft. Therefore, in Table 5, Section 1, find the numbers "33.2-37.2" in the column headed "7-8." This is the re-

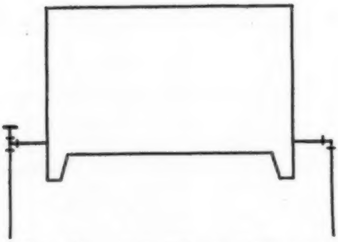

FIG. 5A. LOW FLOW RADIATOR CONNECTION

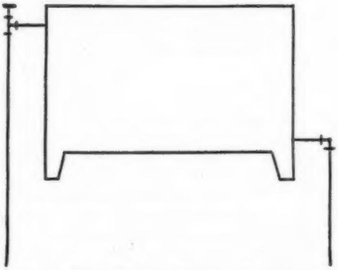

FIG. 5B. HIGH FLOW RADIATOR CONNECTION

quired range and accordingly the orifice size is found on the same line at the extreme left. In this case it is 7/32 in. For radiators 2 and 3 use Table 5, Section 2, and find, in a manner similar that No. 2 (63 sq. ft), requires a 5/16 orifice and No. 3 (22½ sq ft) requires a 13/64 in. orifice. For radiators 4 and 5 use Table 5, Section 3, and find that radiator 4 (31½ sq ft) takes a ¼ in. orifice, and radiator 5 (33¾ sq ft) takes a 9/32 in. orifice. Similarly, using Table 5, Section 4, radiators 6 and 7 (each 57 sq ft) take 7/16 in. orifices.

For the right-hand branch each section is about 16⅔ ft long. Therefore, find radiator 8 in Section 1. Radiator 9 is on the border line between Section 2 and Section 3. In a case like this it is always safer to place the radiator in the section nearest the heater. Therefore, radiator 9 is placed in Section 2. Radiator 10 is in Section 4. Proceeding in the same manner as for the left-hand branch, find that the sizes of the orifices required are as follows:

Radiator 8 (60 sq ft) takes a 9/32 in. orifice (Table 5, Section 1).
Radiator 9 (86¼ sq ft) takes a 13/32 in. orifice (Table 5, Section 2).
Radiator 10 (75 sq ft) takes a ½ in. orifice (Table 5, Section 4).

The design of the system is now complete except for the expansion tank

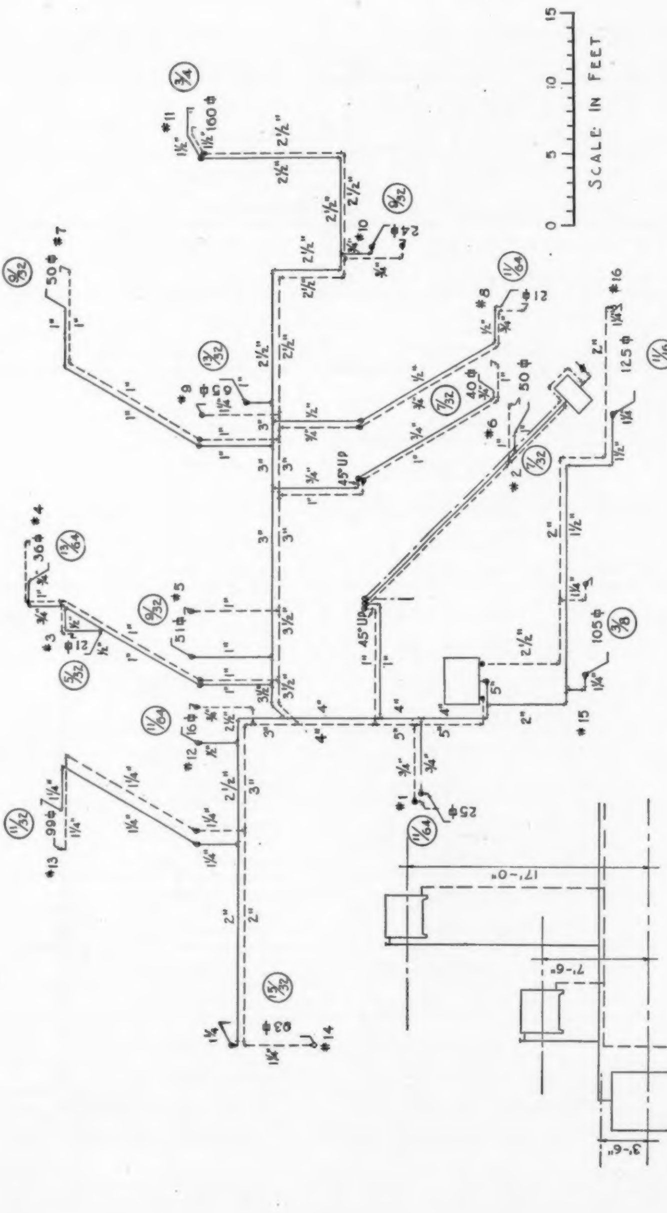

FIG. 6. LAY-OUT FOR A TWO-STORY BUILDING WITH BRANCH MAINS
The sizes of the orifices are indicated by a circle with a number inside

TABLE 6. CAPACITIES IN SQUARE FEET OF DIRECT RADIATION OF ORIFICES DESIGNED FOR VAPOR SYSTEMS

Vapor Rating of Orifice		Section 1													Height Above Midpoint of Heater in Feet														Section 1	Vapor Rating of Orifice	
		4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18		
0	10	7.6	15.2	8.4	16.7	9.0	18.0	9.7	19.4	10.4	20.6	10.9	21.6	11.5	22.6	12.	23.9	12.4	24.5	12.8	25.6	13.3	26.6	13.8	27.6	14.2	28.4	14.6	29.2	0	10
11	20	15.2	22.2	16.7	24.8	18.0	27.0	19.4	28.9	20.6	30.8	21.6	32.5	22.6	33.9	23.9	35.4	24.6	37.	25.6	38.4	26.6	39.8	27.6	41.3	28.4	42.5	29.2	43.7	11	20
21	30	22.2	30.1	24.8	33.2	27.0	35.6	28.9	38.3	30.8	40.7	32.5	43.0	33.9	45.0	35.4	47.5	37.	49.3	38.4	51.2	39.8	53.1	41.3	55.	42.5	56.6	43.7	58.3	21	30
31	40	30.1	37.2	33.2	41.0	35.6	44.2	38.3	48.0	40.7	51.1	43.0	54.0	45.0	56.6	47.5	59.3	49.3	61.5	51.2	64.0	53.1	66.3	55.	68.7	56.6	70.5	58.3	72.8	31	40
41	50	37.2	44.1	41.0	49.4	44.2	53.8	48.0	57.6	51.1	61.4	54.	64.5	56.6	67.9	59 3	71.	61.5	74.0	64.0	76.9	66.3	79.6	68.7	82.5	70.5	85.	72.8	87.6	41	50
51	60	44.1	52.2	49.4	57.5	53.8	62.5	57.6	66.9	61.4	71.0	64.5	75.1	67.9	78.0	71.	82.	74.0	86.1	76.9	89.5	79.6	92.8	82.5	96.	85.	99.	87.6	102.	51	60
61	70	52.2	59.2	57.5	66.5	62.5	71.1	66.9	76.5	71.0	81.0	75.1	85.0	78.0	90.5	82.	94.8	86.1	98.5	89.5	102.5	92.8	106.1	96.	110.	99.	113.	102.	117.	61	70
71	80	59.2	66.6	66.5	73.7	71.1	80.2	76.5	85.0	81.0	92.0	85.	97.2	90.5	102.	94.8	107.	98.5	111.	102.5	115.0	106.1	120.	110.	124.	113.	128.	117.	131.	71	80
81	90	66.6	74.2	73.7	81.0	80.2	87.6	85.0	96.1	92.0	103.	97.2	108.	102.	114.	107.	119.	111.	123.	115.0	128.	120.	133.	124.	138.	128.	142.	131.	146.	81	90
91	100	74.2	81.0	81.0	90.5	87.6	98.3	96.1	105.	103.	112.	108.	119	114.	125.	119.	130.	123.	136.	128	141.	133.	146.	138.	151.	142.	156.	146.	160.	91	100
101	110	81.0	89.3	90.5	98.6	98.3	107.	105.	115.	112.	123.	119.	129	125.	134.	130.	141.	136.	146.	141.	154.	146.	159.	151.	165.	156.	169	160.	175.	101	110
111	120	89.3	97.0	98.6	107.	107.	116.	115.	125.	123.	131.	129.	139	134.	146.	141.	153	146.	160.	154.	167.	159.	173	165.	179.	169.	184	175.	190.	111	120
121	130	97.0	104.	107.	115.	116.	125.	125.	132.	131.	141.	139.	149.	146.	157.	153.	164.	160.	172.	167.	179.	173.	186.	179.	192.	184.	198	190.	204.	121	130
131	140	104.	112.	115.	124.	125.	132.	132.	142.	141.	151.	149.	160.	157.	169.	164.	176.	172.	185	179.	192.	186.	199.	192.	206.	198.	212			131	140
141	150	112.	120.	124.	130.	132.	141.	142.	151.	151.	162.	160.	171.	169.	182.	176.	189.	185.	197	192.	205.	199	212							141	150
151	160	120.	127.	130.	137.	141.	150.	151.	161.	162.	172.	171.	182.	182.	192.	189.	200.	197.	210.											151	160
161	170	127.	131.	137.	146.	150.	159.	161.	171.	172.	182.	182.	193.	192.	202.															161	170
....	....	131.	161.	146.	184.	159.	201.	171.	216.	182.	230.	193.	244.																	....	....
....	....	161.	205.	184.	228.																									....	....
Two	0-10	....	....	....	....	....	....	....	....	....	....	7.6	10.9	7.0	11.5	8.4	12.	8.7	12.4	9.0	12.8	9.3	13.3	9.7	13.8	10.	14.2	10.4	14.6	Two	0-10

Vapor Rating of Orifice		18	19	19	20	20	21	21	22	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60	Vapor Rating of Orifice	
0	10	15.4	30.8	16.2	32.2	16.9	33.8	17.8	35.3	18.1	36.2	18.8	37.6	19.7	39.4	20.3	41.3	21.8	43.6	22.6	45.2	23.4	46.9	24.4	48.9	25.6	51.4	26.8	53.6	0	10
11	20	30.8	46.2	32.2	48.5	33.8	50.6	35.3	52.9	36.2	54.2	37.6	56.2	39.4	59.0	41.3	61.7	43.6	65.2	45.2	67.6	46.9	70.1	48.9	73.2	51.4	76.8	53.6	80.1	11	20
21	30	46.2	61.6	48.5	64.6	50.6	67.6	52.9	70.5	54.2	72.4	56.2	75.0	59.0	78.7	61.7	82.4	65.2	87.	67.6	90.2	70.1	93.5	73.2	97.5	76.8	103.	80.1	107.	21	30
31	40	61.6	77.0	64.6	80.7	67.6	84.5	70.5	88.0	72.4	90.4	75.0	93.6	78.7	98.5	82.4	103.	87.	109.	90.2	113.	93.5	117.	97.5	122.	103.	128.	107.	134.	31	40
41	50	77.0	92.5	80.7	97.	84.5	102.	88.	106.	90.4	109.	93.6	113.	98.5	118.	103.	124.	109.	131.	113.	136.	117.	140.	122.	147.	128.	154.	134.	161.	41	50
51	60	92.5	108.	97.	113.	102.	118.	106.	123.	109.	126.	113.	131.	118.	138.	124.	144.	131.	152.	136.	158.	140.	164.	147.	171.	154.	179.	161.	187.	51	60
61	70	108.	123.	113.	129.	118.	135.	123.	141.	126.	145.	131.	150.	138.	158.	144.	165.	152.	174.	158.	181.	164.	187.	171.	195.	179.	203.	187.	214.	61	70
71	80	123.	139.	129.	146.	135.	152.	141.	159.	145.	163.	150.	169.	158.	177.	165.	186.	174.	196.	181.	204.	187.	211.	195.	220.					71	80
81	90	139.	154.	146.	162.	152.	169.	159.	178.	163.	181.	169.	188.	177.	197.	186.	206.	196.	218.											81	90
91	100	154.	169.	162.	178.	169.	186.	178.	194.	181.	199.	188.	206.	197.	217.															91	100
101	110	169.	185.	178.	193.	186.	202.	194.	211.	199.	217.																			101	110
111	120	185.	201.	193.	211.																									111	120
121	130																													121	130
131	140																													131	140
141	150																													141	150
151	160																													151	160
161	170																													161	170
....	....																													....	....
....	....																													....	....
Two	0-10	10.9	15.4	11.5	16.2	12.0	16.9	12.4	17.8	12.8	18.1	13.3	18.8	14.	19.7	14.6	20.3	15.4	21.8	16.	22.6	16.6	23.4	17.4	24.4	18.1	25.6	18.8	26.8	....	....

Two 0-10 means that two 0-10 orifices are to be placed in the radiators, one in each union.

TABLE 6. CAPACITIES IN SQUARE FEET OF DIRECT RADIATION OF ORIFICES DESIGNED FOR VAPOR SYSTEMS—*Continued*

Vapor Rating of Orifice		Section 2												Height Above Midpoint of Heater in Feet														Section 2		Vapor Rating of Orifice	
		4½	5	5	6	6	7	7	8	8	9	9	10	10	11	11	12	12	13	13	14	14	15	15	16	16	17	17	18		
Two	0–10	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	6.9	9.7	7.2	10.1	7.4	10.5	7.8	10.9	8.	11.2	8.2	11.6	Two	0–10
0	10	5.8	11.6	6.4	13.0	7.1	14.1	7.6	15.1	8.1	16.1	8.6	17.	9.1	18.	9.5	18.8	9.7	19.4	10.	20.2	10.5	21.	10.9	21.8	11.2	22.5	11.6	23.2	0	10
11	20	11.6	17.2	13.0	18.6	14.1	20.4	15.1	22.	16.1	24.1	17.	25.5	18.	26.8	18.8	28.1	19.4	29.1	20.2	30.3	21.	31.4	21.8	32.8	22.5	33.6	23.2	34.7	11	20
21	30	17.2	22.1	18.6	25.5	20.4	27.8	22.	30.	24.1	32.	25.5	33.7	26.8	35.6	28.1	37.6	29.1	38.9	30.3	40.4	31.4	42.	32.8	43.4	33.6	44.9	34.7	46.3	21	30
31	40	22.1	28.4	25.5	31.0	27.8	34.1	30.	37.	32.	39.5	33.7	42.	35.6	43.9	37.6	46.	38.9	48.5	40.4	50.5	42.	52.4	43.4	54.2	44.9	56.	46.3	57.8	31	40
41	50	28.4	33.4	31.0	37.4	34.1	41.	37.	43.8	39.5	47.	42.	50.	43.9	52.5	46.	55.6	48.5	58.2	50.5	60.6	52.4	63.	54.2	65.1	56.	67.4	57.8	69.5	41	50
51	60	33.4	38.8	37.4	42.5	41.	48.3	43.8	53.6	47.	55.6	50.	59.	52.5	62.9	55.6	64.7	58.2	67.9	60.6	70.7	63.	73.4	65.1	75.9	67.4	78.5	69.5	81.	51	60
61	70	38.8	44.5	42.5	50.	48.3	55.1	53.6	59.5	55.6	63.2	59.	67.	62.9	71.	64.7	74.4	67.9	77.6	70.7	80.8	73.4	83.8	75.9	86.8	78.5	89.6	81.	92.5	61	70
71	80	44.5	50.8	50.	56.8	55.1	62.3	59.5	66.6	63.2	71.3	67.	75.6	71.	80.	74.4	82.8	77.6	87.5	80.8	91.	83.8	94.5	86.8	97.7	89.6	101.	92.5	104.	71	80
81	90	50.8	56.5	56.8	63.	62.3	68.5	66.6	74.1	71.3	79.2	75.6	82.6	80.	90.3	82.8	93.5	87.5	97.3	91.	101.	94.5	105.	97.7	109.	101.	112.	104.	116.	81	90
91	100	56.5	62.	63.	68.6	68.5	75.3	74.1	81.	79.2	87.7	82.6	93.1	90.3	98.	93.5	103.	97.3	107.	101.	111.	105.	115.	109.	119.	112.	123.	116.	127.	91	100
101	110	62.	66.6	68.6	75.	75.3	81.	81.	89.6	87.7	95.7	93.1	102.	98.	107.	103.	112.	107.	117.	111.	121.	115.	126.	119.	131.	123.	135.	127.	139.	101	110
111	120	66.6	69.4	75.	78.5	81.	89.6	89.6	98.5	95.7	104.	102.	110.	107.	116.	112.	122.	117.	126.	121.	131	126.	136.	131.	141.	135.	146.	139.	151.	111	120
121	130	69.4	74.5	78.5	87.9	89.6	96.3	98.5	104.	104.	111.	110.	118.	116.	124.	122.	130.	126.	136.	131.	141.	136.	147.	141.	152.	146.	157.	151.	162.	121	130
131	140	74.5	81.	87.9	90.6	96.3	100.	104.	108.	111.	116.	118.	123.	124.	129.	130.	133.	136.	145.	141.	152.	147.	157.	152.	163.	157.	168.	162.	174.	131	140
141	150	81.	90.1	90.6	100.1	100.	110.	108.	119.	116.	127.	123.	133.	129.	140.	133.	147.	145.	155.	152.	162.	157.	168.	163.	174.	168.	179.	174.	185.	141	150
151	160	90.1	96.	100.1	106.5	110.	117.	119.	127.	127.	132.	133.	141.	140.	149	147.	156.	155.	165.	162.	172.	168.	179.	174.	185.	179.	191.	185.	197.	151	160
161	170	96.	101.5	106.5	112.	117.	124.	127.	131.	132.	140.	141.	149.	149.	158.	156.	166.	165.	175.	172.	182.	179.	189.	185.	196.	191.	202.	197.	209.	161	170
½	....	101.5	122.2	112.	132.	124.	145.	131.	158.	140.	170.	149.	180.	158.	191.	166.	200.	175.	221.	182.	230.	189.	239.	196.	247.	202.	....	209.	....	½	....
9/16	....	122.2	151.	132.	181.	145.	180.	158.	204.	170.	219.	180.	233.	191.	246.	200.	....	221.	....	230.	....	239.	....	247.	....	....	....	....	....	9/16	....
5/8	....	151.	166.2	171.	188.	180.	207.	204.	....	219.	....	233.	....	246.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	5/8	....
11/16	....	166.	293.	188.	330.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11/16	....

		18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60		
Two	0–10	8.7	12.3	9.1	12.9	9.6	13.5	10.	14.1	10.3	14.6	10.7	15.3	11.3	15.9	11.8	16.9	12.4	17.6	12.9	18.4	13.4	18.9	14.	19.8	14.7	20.7	15.3	21.6	Two	0–10
0	10	12.3	24.6	12.9	25.8	13.5	27.	14.1	28.2	14.6	29.2	15.3	30.4	15.9	31.8	16.9	33.8	17.6	35.2	18.3	36.6	18.9	37.8	19.8	39.5	20.7	41.5	21.6	43.3	0	10
11	20	24.6	36.8	25.8	38.6	27.	40.5	28.2	42.7	29.2	43.9	30.4	45.5	31.8	47.7	33.8	50.5	35.2	52.6	36.6	55.7	37.8	56.6	39.5	59.1	41.5	62.	43.3	64.9	11	20
21	30	36.8	49.	38.6	51.5	40.5	54.	42.7	56.2	43.9	58.5	45.5	61.6	47.7	63.6	50.5	67.5	52.6	70.2	55.7	73.	56.6	75.5	59.1	78.9	62.	82.6	64.9	86.5	21	30
31	40	49.	61.1	51.5	64.3	54.	67.3	56.2	70.2	58.5	73.	61.6	75.6	63.6	79.5	67.5	84.2	70.2	87.6	73.	91.2	75.5	94.3	78.9	98.5	82.6	103.	86.5	108.	31	40
41	50	61.1	73.5	64.3	77.2	67.3	81.	70.2	84.4	73.	87.6	75.6	91.	79.5	95.5	84.2	101.	87.6	105.	91.2	110.	94.3	113.	98.5	118.	103.	124.	108.	130.	41	50
51	60	73.5	85.5	77.2	90.	81.	94.2	84.4	99.3	87.6	102.	91.	106.	95.5	111.	101.	118.	105.	123.	110.	127.	113.	132.	118.	138.	124.	145.	130.	151.	51	60
61	70	85.5	98.	90.	103.	94.2	108.	99.3	112.	102.	117.	106.	121.	111.	127.	118.	133.	123.	140.	127.	146.	132.	151.	138.	158.	145.	165.	151.	173.	61	70
71	80	98.	110.	103.	116.	108.	122.	112.	127.	117.	132.	121.	137.	127.	143.	135.	152.	140.	158.	146.	165.	151.	170.	158.	178.	165.	186.	173.	195.	71	80
81	90	110.	123.	116.	129.	122.	135.	127.	141.	132.	146.	137.	152.	143.	159.	152.	169.	158.	176.	165.	183.	170.	189.	178.	198.	186.	207.	195.	216.	81	90
91	100	123.	135.	129.	142.	135.	148.	141.	151.	146.	160.	152.	167.	159.	175.	169.	185.	176.	193.	183.	205.	189.	208.	198.	217.	207.	....	216.	....	91	100
101	110	135.	147.	142.	154.	148.	162.	151.	169.	160.	175.	167.	182.	175.	191.	185.	202.	193.	211.	205.	....	208.	....	217.	....	....	....	....	....	101	110
111	120	147.	159.	154.	167.	162.	176.	169.	183.	175.	190.	182.	197.	191.	207.	202.	....	211.	....	....	....	....	....	....	....	....	....	....	....	111	120
121	130	159.	171.	167.	180.	176.	188.	183.	197.	190.	204.	197.	212.	207.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	121	130
131	140	171.	184.	180.	193.	188.	202.	197.	211.	204.	....	212.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	131	140
141	150	184.	196.	193.	206.	202.	....	211.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	141	150
151	160	196.	210.	206.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	151	160
161	170	210.	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	161	170
½	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	½	....
9/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	9/16	....
5/8	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	5/8	....
11/16	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11/16	....

TABLE 6. CAPACITIES IN SQUARE FEET OF DIRECT RADIATION OF ORIFICES DESIGNED FOR VAPOR SYSTEMS—*Continued*

Vapor Rating of Orifice		Section 3									Height Above Midpoint of Heater in Feet																Section 3	Vapor Rating of Orifice	
		4½	5	5	6	6	7	7	8	8	9	9	10	10	11	12	12	12	13	13	14	14	15	15	16	16	17	17	18		
Two	0-10																													Two	0-10
0	10	4.2	8.5	4.8	9.5	5.2	10.5	5.7	11.4	6.1	12.2	6.5	13.0	6.8	13.7	7.2	14.4	7.3	14.6	7.6	15.2	8.	15.8	8.2	16.5	8.5	17.1	8.8	17.6	0	10
11	20	8.5	12.7	9.5	14.3	10.5	15.3	11.4	16.7	12.2	18.0	13.0	19.2	13.7	20.2	14.4	21.0	14.6	21.9	15.2	22.8	15.8	23.8	16.5	24.7	17.1	25.5	17.6	26.3	11	20
21	30	12.7	16.3	14.3	18.4	15.3	20.4	16.7	21.5	18.0	24.0	19.2	25.6	20.2	27.0	21.0	28.4	21.9	29.1	22.8	30.4	23.8	31.6	24.7	32.9	25.5	34.	26.3	35.1	21	30
31	40	16.3	20.2	18.4	22.0	20.4	25.6	21.5	27.8	24.0	29.9	25.6	31.7	27.0	33.2	28.4	34.9	29.1	36.4	30.4	38.0	31.6	39.6	32.9	41.1	34.	42.5	35.1	43.8	31	40
41	50	20.2	24.4	22.0	27.7	25.6	30.6	27.8	33.2	29.9	35.1	31.7	37.5	33.2	39.8	34.9	42.0	36.4	43.6	38.0	45.6	39.6	47.5	41.1	49.4	42.5	51.	43.8	52.6	41	50
51	60	24.4	28.4	27.7	32.1	30.6	34.8	33.2	38.0	35.1	41.9	37.5	43.6	39.8	44.6	42.0	49.2	43.6	50.9	45.6	53.1	47.5	55.4	49.4	57.5	51.	59.5	52.6	61.4	51	60
61	70	28.4	32.5	32.1	35.7	34.8	39.7	38.0	43.5	41.9	47.4	43.6	50.5	44.6	53.5	49.2	56.4	50.9	58.2	53.1	60.9	55.5	63.4	57.5	65.6	59.5	68.	61.4	70.2	61	70
71	80	32.5	34.9	35.7	40.1	39.7	43.6	43.5	49.5	47.4	53.4	50.5	56.9	53.5	60.2	56.4	63.2	58.2	65.6	60.9	69.5	63.4	71.4	65.6	74.0	68.	76.6	70.2	78.	71	80
81	90	34.9	38.8	40.1	43.6	43.6	50.4	49.5	55.0	53.4	59.2	56.9	63.2	60.2	66.4	63.2	69.8	65.6	73.0	69.5	76.1	71.4	79.4	74.0	82.4	76.6	85.1	78.	88.	81	90
91	100	38.8	42.6	43.6	49.6	50.4	55.4	55.0	60.3	59.2	64.1	63.2	68.5	66.4	72.6	69.8	76.5	73.0	80.0	76.1	83.6	79.4	87.0	82.4	90.4	85.1	93.5	88.	96.	91	100
101	110	42.6	47.8	49.6	54.4	55.4	60.5	60.3	65.1	64.1	70.2	68.5	75.0	72.6	79.5	76.5	83.5	80.0	87.4	83.6	91.2	87.0	95.0	90.4	98.5	93.5	102.	96.5	105.	101	110
111	120	47.6	51.5	54.4	58.7	60.5	64.1	65.1	70.2	70.2	75.8	75.0	81.0	79.5	83.5	83.5	91.0	87.4	94.6	91.2	99.0	95.0	106.	98.5	107.	102.	111.	105.	114.	111	120
121	130	51.5	55.5	58.7	63.2	64.1	68.1	70.2	75.8	75.8	81.0	81.0	88.2	83.5	93.4	91.0	98.0	94.6	104	99.0	106.	106.	111.	107.	115.	111.	119.	114.	123.	121	130
131	140	55.5	59.5	63.2	66.5	68.1	74.1	75.8	81.0	81.0	88.6	88.2	94.5	93.4	100.	98.0	105.	104.	109.	106.	114.	111.	119.	115.	123.	119.	127.	123.	131.	131	140
141	150	59.5	63.2	66.5	71.0	74.1	79.2	81.0	87.9	88.6	94.5	94.5	101.	100.	107.	105.	112.	109.	116.	114.	122.	119.	127.	123.	132.	127.	136.	131.	140.	141	150
151	160	63.2	65.5	71.0	75.5	79.2	82.5	87.9	93.4	94.5	100.	101.	107.	107.	113.	112.	116.	116.	124.	122.	129.	127.	135.	132.	140.	136.	145.	140.	149.	151	160
161	170	65.5	69.5	75.5	80.0	82.5	84.8	93.4	99.0	100.	106.	107.	113.	113.	120.	116.	126.	124.	131.	129.	137.	135.	143.	140.	148.	145.	153.	149.	158.	161	170
½	....	69.5	90.1	80.0	103.	84.8	115.	99.0	125.	106.	131.	113.	139.	120.	147.	126.	151.	131.	166.	137.	173.	143.	180.	148.	187.	153.	194.	158.	200.	½	....
9/16	....	90.1	111.	103.	127.	115.	134.	125.	147.	131.	157.	139.	172.	147.	183.	151.	194.	166.	205.	173.	214.	180.	222.	187.	231	194.	239.	200.		9/16	....
5/8	....	111.	123.	127.	143.	134.	162.	147.	178.	157.	193.	172.	207.	183.	220.	194.	232.	205.		214.		222.		231.		239.				5/8	....
11/16	....	123.	145.	143.	172.	162.	193.	178.	213.	193.	230.																			11/16	....
¾	....	145.	200.	172.	234.	193.	263.																							¾	....

		18	20	20	22	22	24	24	26	26	28	28	30	30	33	33	36	36	40	40	43	43	46	46	50	50	55	55	60		
Two	0-10			7.	9.8	7.3	10.3	7.6	10.8	7.9	11.2	8.2	11.6	8.7	12.2	9.1	12.8	9.6	13.5	10.	14.1	10.3	14.6	10.8	15.2	11.3	16.	13.5	16.8	Two	0-10
0	10	9.3	18.6	9.8	19.6	10.3	20.6	10.8	21.5	11.2	22.4	11.6	23.2	12.2	24.3	12.8	25.6	13.5	27.1	14.1	28.8	14.6	29.2	15.2	30.4	16.	32.	16.8	33.5	0	10
11	20	18.6	27.9	19.6	29.4	20.8	30.9	21.5	32.3	22.4	33.5	23.2	34.8	24.3	36.6	25.6	38.4	27.1	40.7	28.2	42.1	29.2	43.7	30.4	45.7	32.	48.	33.5	50.1	11	20
21	30	27.9	37.2	29.4	39.2	30.9	41.2	32.3	43.8	33.5	44.6	34.8	46.5	36.6	48.8	38.4	51.2	40.7	54.1	42.1	56.2	43.7	58.2	45.7	60.8	48.	63.8	50.1	66.9	21	30
31	40	37.2	46.4	39.2	49.	41.2	51.5	43.8	53.8	44.6	55.8	46.5	58.0	48.8	61.	51.2	63.8	54.1	67.6	56.2	70.2	58.2	72.6	60.8	76.	63.8	79.7	66.9	83.5	31	40
41	50	46.4	55.7	49.	59.	51.5	61.8	53.8	64.6	55.8	67.0	58.0	69.6	61.	73.3	63.8	76.7	67.6	81.2	70.2	84.4	72.6	86.4	76.	91.3	79.7	95.	83.5	100.	41	50
51	60	55.7	65.	59.	68.5	61.8	72.	64.6	75.2	67.	78.	69.6	81.1	73.3	85.4	76.7	89.4	81.2	94.5	84.4	98.1	86.4	102.	91.3	106.	95.	112.	100.	117.	51	60
61	70	65.	74.4	68.5	78.4	72.	82.4	75.2	86.0	78.	89.4	81.1	92.8	85.4	97.6	89.4	102.	94.5	108.	98.1	112.	102.	116.	106.	122.	112.	128.	117.	134.	61	70
71	80	74.4	83.6	78.4	83.3	82.4	92.8	86.	97.	89.4	100.	92.8	105.	97.6	110.	102.	115.	108.	122.	112.	127.	116.	131.	122.	137.	128.	143.	134.	151.	71	80
81	90	83.6	93.	88.3	98.1	92.8	103.	97.	108.	100.	112.	105.	116.	110.	122.	115.	128.	122.	136.	127.	141.	131.	146.	137.	152.	143.	160.	151.	168.	81	90
91	100	93.	102.	98.1	108.	103.	113.	108.	118.	112.	123.	116.	127.	122.	134.	128.	141.	136.	150.	141.	154.	146.	160.	152.	167.	160.	176.	168.	184.	91	100
101	110	102.	112.	108.	117.	113.	124.	118.	129.	123.	134.	127.	139.	134.	147.	141.	154.	150.	162.	154.	169.	160.	175.	167.	182.	176.	191.	184.	200.	101	110
111	120	112.	121.	117	127.	124.	134.	129.	140.	134.	145.	139.	151.	147.	159.	154.	166.	162.	176.	169.	183.	175.	189.	182.	198.	191.	208.	200.		111	120
121	130	121.	130.	127.	137.	134.	144.	140.	150.	145.	156.	151.	162.	159.	171.	166.	179.	176.	189.	183.	196.	189.	204.	198.	212.	208.				121	130
131	140	130.	139.	137.	147.	144.	155.	150.	161.	156.	168.	162.	174.	171.	183.	179.	192.	189.	203.	196.	211.	204.		212.						131	140
141	150	139.	149.	147.	157.	155.	165.	161.	172.	168.	179.	174.	185.	183.	195.	192	204.	203.		211.										141	150
151	160	149.	158.	157.	167.	165.	175.	172.	183.	179.	190.	185.	199.	195.	208.	204.														151	160
161	170	158.	167.	167.	177.	175.	185.	183.	194.	190.	201.	199.	209.	208.																161	170
½	....	167.	212.	177.	223.	185.	234.	194.	245.	201.		209.																		½	....
9/16	....	212.		223.		234.		245.																						9/16	....
5/8	....																													5/8	....
11/16	....																													11/16	....
¾	....																													¾	....

TABLE 6. CAPACITIES IN SQUARE FEET OF DIRECT RADIATION OF ORIFICES DESIGNED FOR VAPOR SYSTEMS—*Continued*

Vapor Rating of Orifice	Section 4						Height Above Midpoint of Heater in Feet							Secton 4.	Vapor Rating of Orifice
	4½—5	5—6	6—7	7—8	8—9	9—10	10—11	11—12	12—13	13—14	14—15	15—16	16—17	17—18	
0-10	2.9- 5.8	3.3- 6.7	3.7- 7.4	4.0- 8.1	4.3- 8.7	4.6- 9.3	4.9- 9.8	5.1- 10.4	5.2- 10.5	5.3- 10.7	5.5- 10.9	5.7- 11.4	5.9- 11.9	6.1- 12.3	0-10
11-20	5.8- 8.7	6.7- 10.0	7.4- 11.1	8.1- 12.1	8.7- 13.0	9.3- 13.9	9.8- 14.7	10.4- 15.0	10.5- 15.2	10.7- 15.6	10.9- 16.3	11.4-17.01	11.9- 17.7	12.3- 18.4	11-20
21-30	8.7- 10.7	10.0- 13.3	11.1- 14.8	12.1- 15.3	13.0- 16.5	13.9- 17.7	14.7- 18.9	15.0- 20.0	15.2- 20.2	15.6- 20.8	16.3- 21.8	17.01-22.8	17.7- 23.6	18.4- 24.5	21-30
31-40	10.7- 14.5	13.3- 15.4	14.8- 17.2	15.3- 19.0	16.5- 20.6	17.7- 21.0	18.9- 22.6	20.0- 25.6	20.2- 24.7	20.8- 25.9	21.8- 27.2	22.8- 28.4	23.6- 29.5	24.5- 30.6	31-40
41-50	14.5- 15.5	15.4- 18.1	17.2- 20.7	19.0- 21.3	20.6- 23.4	21.0- 26.8	22.6- 28.3	25.6- 30.0	24.7- 30.3	25.9- 31.2	27.2- 32.7	28.4- 34.0	29.5- 35.4	30.6- 36.8	41-50
51-60	15.5- 18.1	18.1- 21.0	20 7- 22.1	21.3- 26.4	23.4- 28.8	26.8- 31.0	28.3- 33.0	30.0- 33.8	30.3- 34.7	31.2- 36.4	32.7- 38.1	34.0- 40.2	35.4- 41.3	36.8- 42.9	51-60
61-70	18.1- 20.7	21.0- 24.2	22.1- 27.6	26 4- 32.0	28.8- 33.0	31.0- 34.1	33.0- 36.4	33.8- 38.6	34.7- 39.6	36.4- 41.7	38.1- 43.5	40.2- 45.4	41.3- 47.3	42.9- 49.0	61-70
71-80	20.7- 23.4	24.2- 27.4	27.6- 31.0	32.0- 33.2	33.0- 35.6	34.1- 38.4	36.4- 41.0	38.6- 43.6	39.6- 44.7	41.7- 47.0	43.5- 49.0	45.4- 51.2	47.3- 53.3	49.0- 55.2	71-80
81-90	23.4- 26.0	27.4- 30.6	31.0- 33.2	33.2- 36.2	35.6- 39.6	38.4- 42.7	41.0- 44.4	43.6- 47.0	44.7- 49.7	47.0- 52.3	49.0- 54.5	51.2- 57.0	53.3- 59.2	55.2- 61.4	81-90
91-100	26.0- 28.4	30.6- 33.2	33.2- 35.5	36.2- 39.6	39.6- 43.5	42.7- 45.2	44.4- 51.0	47.0- 54.0	49.7- 54.5	52.3- 57.4	54.5- 59.7	57.0- 62.4	59.2- 65.0	61.4- 67.4	91-100
101-110	28.4- 31.0	33.2- 33.6	35.5- 38.7	39.6- 43.4	43.5- 45.8	45.2- 52.1	51.0- 55.6	54.0- 58.9	54.5- 59.4	57.4- 62.6	59.7- 65.2	62.4- 68.0	65.0- 71.0	67.4- 73.5	101-110
111-120	31.0- 33.0	33.6- 36.4	38.7- 41.9	43.4- 44.5	45.8- 52.6	52.1- 56.6	55.6- 60.4	58.9- 63.2	59.4- 64.5	62.6- 68.0	65.2- 71.0	68.0- 73.8	71.0- 77.0	73.5- 80.0	111-120
121-130	33.0- 34.0	36.4- 39.1	41.9- 43.6	44.5- 51.9	52.6- 56.5	56.6- 60.8	60.4- 63.5	63.2- 67.4	64.5- 69.3	68.0- 73.0	71.0- 76.0	73.8- 79.4	77.0- 82.6	80.0- 85.6	121-130
131-140	34.0- 35.0	39.1- 41.8	43.6- 45.4	51.9- 55.4	56.5- 60.5	60.8- 63.6	63.5- 68.1	67.4- 72.6	69.3- 74.3	73.0- 78.8	76.0- 81.4	79.4- 85.0	82.6- 88.7	95.6- 91.9	131-140
141-150	35.0- 36.6	41.8- 43.6	45.4 53.4	55.4- 59.3	60.5- 63.2	63.6- 68.0	68.1- 72.6	72.6- 77.1	74.3- 79.2	78.0- 83.5	81.4- 87.0	85.0- 91.0	88.7- 94.5	91.9- 98.0	141-150
151-160	36.6- 38.8	43.6- 45.0	53.4- 55.0	59.3- 58.4	63.2- 64.4	68.0- 70.0	72.6- 75.2	77.1- 80.0	79.2- 84.3	83.5- 88.0	87.0- 92.0	91.0- 96.0	94.5-101.	98.0-106.	151-160
161-170	38.8- 41.2	45.0- 46.1	55.0- 60.1	58.4- 64.5	64.4- 70.8	70.0- 76.4	75.2- 81.9	80.0- 82.3	84.3- 89.4	88.0- 94.0	92.0- 98.0	96.0- 102.	101.-106.	106.-110.	161-170
½"	41.2- 45.2	46.1- 64.0	60.1- 72.8	64.5- 81.5	70.-8 91.8	76.4- 98.5	81.9-105.	82.3-111.	89.4-113.	94.0-118.	98.0-124.	102. -131.	106.- 137.5	110.-143.	½"
9/16"	45.2- 63.2	64.0- 77.6	72.8- 79.0	81.5-104.	91.8-113.	98.5-123.	105. -130.	111. -136.	113. -138.1	118. -147.	124. -153.	131. -160.	137.5-166.	143. -172.	9/16"
⅝"	63.2- 76.2	77.6- 95.2	79.0-112.	104. -125.	113. -136.	123. -137.	130. -146.	136. -157.	138.1-168.	147. -177.	153. -184.	160. -193.	166. -201.	172. -208.	⅝"
11/16"	76.2- 97.2	95.2-117.	112. -134.	125. -133.	136. -148.	137. -163.	146. -176.	157. -187.	168. -201.	177. -211.	184. -221.	193. -23.	....	....	11/16"
¾"	97.2-132.	117. -131.	134. -158.	133. -184.	148. -201.	163. -220.	176. -238.	187. -254.	....	....	....	....	....	....	¾"
⅞"	132.- 152.	131. -191.	158. -210.	184. -240.	....	....	....	....	....	....	....	....	....	....	⅞"
1"	152.- 191.	191. -252.	....	....	....	....	....	....	....	....	....	....	....	....	1"

	18—20	20—22	22—24	24—26	26—28	28—30	30—33	33—36	36—40	40—43	43—46	46—50	50—55	55—60	
0-10	6.5- 13.1	6.9- 13.8	7.3- 14.6	7.7- 15.3	8.0- 16.0	8.3- 16.5	8.8- 17.5	9.2- 18.4	9.8- 19.6	10.1- 20.7	10.5- 21.0	11.1- 22.0	11.6- 23.1	12.1- 24.1	0-10
11-20	13.1- 19.6	13.8- 20.7	14.6- 21.8	15.3- 23.0	16.0- 23.9	16.5- 24.8	17.5- 26.1	18.4- 27.4	19.6- 29.2	20.7- 30.2	21.0- 31.4	22.0- 33.0	23.1- 34.6	24.1- 36.1	11-20
21-30	19.6-26.5	20.7- 27.6	21.8- 29.1	23.0- 30.6	23.9- 31.9	24.8- 33.0	26.1- 35.0	27.4- 36.6	29.2- 39.0	30.2- 40.5	31.4- 42.0	33.0- 44.0	34.6- 46.1	36.1- 48.2	21-30
31-40	26.5-32.5	27.6- 35.5	29.1- 36.4	30.6- 38.2	31.9- 39.8	33.0- 41.3	35.0- 43.5	36.6- 45.7	30.0- 48.7	40.5- 50.5	42.0- 52.3	44.0- 54.8	46.1- 57.6	48.2- 60.2	31-40
41-50	32.5- 39.2	35.5- 41.5	36.4- 43.8	38.2- 45.8	39.8- 48.0	41.3- 49.5	43.5- 52.3	45.7- 55.0	48.7- 58.5	50.5- 60.5	52.3- 63.0	54.8- 66.0	57.6- 69.2	60.2- 72.5	41-50
51-60	39.2- 45.6	41.5- 48.3	43.8- 51.0	45.8- 53.4	48.0- 55.7	49.5- 57.7	52.3- 61.0	55.0- 64.0	58.5- 68.1	60.5- 70.6	63.0- 73.3	66.0- 76.9	69.2- 80.5	72.5- 84.5	51-60
61-70	45.6- 52.1	48.3- 55.4	51.0- 58.2	53.4- 61.0	55.7- 63.8	57.7- 66.0	61.0- 68.6	64.0- 7.22	68.1- 76.0	70.6- 80.7	73.3- 84.0	76.9- 87.8	80.5- 92.3	84.5- 97.2	61-70
71-80	52.1- 58.7	55.4- 62.3	58.2- 65.5	61.0- 68.9	63.8- 72.0	66.0- 74.5	68.6- 76.1	72.2- 82.5	76.0- 87.8	80.7- 91.0	84.0- 94.5	87.8- 99.0	92.3-104.	97.2-109.	71-80
81-90	58.7- 65.4	62.3- 69.3	65.5- 73.0	68.9- 76.5	72.0- 80.0	74.5- 82.9	76.1- 85.0	82.5- 91.8	87.8- 97.5	91.0-101.0	94.5-105.	99.0-110.	104. -116.	109. -121.	81-90
91-100	65.4- 71.6	69.3- 76.0	73.0- 80.0	76.5- 84.5	80.0- 87.6	82.9- 91.0	85.0- 96.0	91.8-101.1	97.5-107.	101.0-111.	105. -115.6	110. -121.	116. -127.	121. -133.	91-100
101-110	71.6- 78.2	76.0- 83.0	80.0- 87.3	84.5- 91.6	87.6- 95.6	91.0- 99.1	96.0-104.	101.1-110.	107. -117.	111. -121.	115.6-126.	121. -128.	127. -136.	133. -145.	101-110
111-120	78.2- 85.0	83.0- 90.0	87.3- 94.8	91.6- 99.5	95.6-104.	99.1-109.	104. -114.	110. -119.	117. -126.	121. -131.	126. -136.	128. -143.	136. -150.	145. -157.	111-120
121-130	85.0- 91.0	90.0- 96.8	94.8-101.9	99.5-107.	104. -112.	109. -116.	114. -121.	119. -128.	126. -135.	131. -141.	136. -147.	143. -154.	150. -161.	157. -169.	121-130
131-140	91.0- 97.9	96.8-103.8	101.9-109.	107. -114.	112. -119.	116. -124.	121. -130.	128.-137.2	135. -146.	141. -151.	147. -158.	154. -165.	161. -173.	169. -181.	131-140
141-150	97.9-104.	103.8-110.	109. -116.	114. -122.	119. -128.	124. -132.	130. -140.	137.2-147.	146. -155.	151. -161.	158. -168.	165. -175.	173. -184.	181. -193.	141-150
151-160	104. -111.	110. -118.	116. -124.	122. -130.	128. -135.	132. -140.	140. -148.	147. -155.	155. -163.	161. -171.	168. -179.	175. -187.	184. -196.	193. -205.	151-160
161-170	111. -117.	118. -124.	124. -131.	130. -137.	135. -144.	140. -148.	149. -156.	155. -165.	166. -173.	171. -181.	179. -189.	187. -198.	196. -207	....	161-170
½"	117. -150.	124. -158.	131. -166	137. -174.	144. -182.	149.- 190.	156. -200.	165. -210.	173. -.220	181. -229.	189. -239.	....	....	....	½"
9/16"	150. -183.	158. -194.	166. -202.	174. -214.	132. -224.	190. -231.	....	....	....	....	....	....	....	....	9/16"
⅝"	183. -220.	194. -235.	....	....	....	....	....	....	....	....	....	....	....	....	⅝"
11/16"	....	....	....	....	....	....	....	....	....	....	....	....	....	....	11/16"
¾"	....	....	....	....	....	....	....	....	....	....	....	....	....	....	¾"
⅞"	....	....	....	....	....	....	....	....	....	....	....	....	....	....	⅞"
1"	....	....	....	....	....	....	....	....	....	....	....	....	....	....	1"

which is of the usual size and installed in the usual manner. The sizes of the orifices are indicated on the plan by a circle with a number inside.

HOW TO SELECT ORIFICE SIZES WHEN ORIFICES ARE RATED IN SQUARE FEET OF VAPOR RADIATION

If, instead of making the orifices on the job, it is preferable that valves be purchased already equipped with orifices having their capacity in square feet of vapor radiation marked on them, the procedure is exactly the same except that

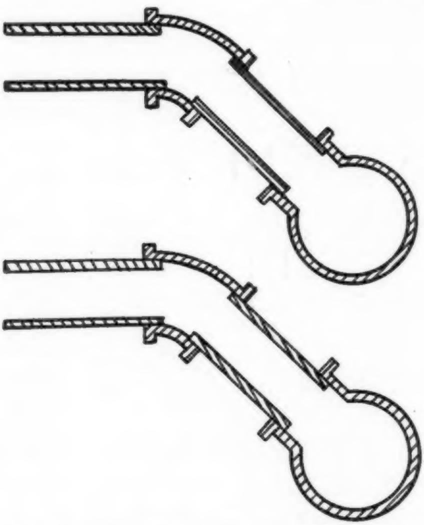

FIG. 7. SUGGESTED VERTICAL SECTION OF MAINS AND RISER LATERALS FOR LINES HAVING BRANCH MAINS

Table 6 is used instead of Table 5. When this is done, the following sizes are obtained:

Radiator 1 (Section 1, 35 sq ft) takes a 21-30 orifice.
Radiator 2 (Section 2, 63 sq ft) takes a 71-80 orifice.
Radiator 3 (Section 2, 22½ sq ft) takes a 21-30 orifice.
Radiator 4 (Section 3, 31½ sq ft) takes a 41-50 orifice.
Radiator 5 (Section 3, 33¾ sq ft) takes a 51-60 orifice.
Radiator 6 (Section 4, 57 sq ft) takes a 141-150 orifice.
Radiator 7 (Section 4, 57 sq ft) takes a 141-150 orifice.
Radiator 8 (Section 1, 60 sq ft) takes a 41-50 orifice.
Radiator 9 (Section 2, 86¼ sq ft) takes a 101-110 orifice.
Radiator 10 (Section 4, 75 sq ft) takes a ½ in. orifice.

It will be noticed that a few inch sizes are included with the vapor sizes. This is because a large first-floor radiator may, in some cases, require a larger

orifice than is made for vapor systems. These extra large sizes will, of course, have to be made on the job.

Settings for Adjustable Orifice Valves

So far as the writer knows, these valves are made only in the ¾ in. size. This means that it will often be necessary to reduce the size of the flow risers in order to connect the valve. This will probably not cause any serious trouble, if the reduction is made fairly close to the valve. It is, of course, especially important to use a full-sized return union elbow. The number of radiator sizes available is also somewhat more limited than in the two previous cases, but the range is large enough so that these valves work all right on some systems. In determining the valve setting, use Table 7 which differs from Tables 5 and 6 in that the number of square feet of radiation is given in the margin and the settings comprise the body of the table.

To determine the setting for radiator 1, locate in Table 7, Section 1, the column headed "7-8" and find on the line with 35 the number "36," which is the required setting. The setting for radiator 2, may be selected from the column headed "7-8" in Table 7, Section 2. The number 63, however, is not located in the margin of the table, so by interpolation, find that the required setting is about 85.

Proceeding in a similar manner, the valve settings may all be determined as follows:

Radiator 1 (Section 1, 35 sq ft) the setting is 36.
Radiator 2 (Section 2, 63 sq ft) the setting is 85.
Radiator 3 (Section 2, 22½ sq ft) the setting is 30.
Radiator 4 (Section 3, 31½ sq ft) the setting is 55.
Radiator 5 (Section 3, 33¾ sq ft) the setting is 62.
Radiator 6 (Section 4, 57 sq ft) the setting is 155.
Radiator 7 (Section 4, 57 sq ft) the setting is 155.
Radiator 8 (Section 1, 60 sq ft) the setting is 58.
Radiator 9 (Section 2, 86¼ sq ft) the setting is 114.
Radiator 10 (Section 4, 75 sq ft) the setting is 220.[2]

How to Deal With Branched Circuits

In Fig. 6, page 263, is illustrated a larger system than the one just considered. It differs from it in that it has second-floor radiators and a branched circuit. In the case of a branched circuit of this sort it is probably best to arrange the mains one above the other with the flow mains on top, as illustrated in Fig. 7, instead of side by side as is the usual practice. If this is not practical, the shorter branch may be carried over the longer one by means of two 45 deg bends as illustrated in Fig. 8, page 247. If head room is important, as it usually is, the *Y* connections may be rotated downward until the upper pipes lie flat upon the lower pipes.

It will also be noticed that in some cases the construction of the building is such that the lateral portions of the risers require an extra turn. In cases like this, illustrated by radiators 2 and 6, it is best to go up with a 45 deg elbow instead of a 90 deg elbow, as illustrated by Fig. 9, page 277, or else make the turn with a 45 deg elbow as illustrated in Fig. 10. In either case, use one 90 deg elbow and one 45 deg elbow, where two 90 deg elbows would other-

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[2] If the valve is calibrated only up to 200 this will be near enough.

TABLE 7—SETTING OF VALVES CALIBRATED IN SQUARE FEET OF RADIATION

Square Feet of Radiation	Section 1													Height Above Midpoint of Heater in Feet												Section 1			Square Feet of Radiation
	4½-5	5-6	6-7	7-8	8-9	9-10	10-11	11-12	12-13	13-14	14-15	15-16	16-17	17-18	18-20	20-22	22-24	24-26	26-28	28-30	30-33	33-36	36-40	40-43	43-46	46-50	50-55	55-60	
10	13	12	11	10	10	9	9																						
15	20	18	16	15	14	14	13	13	12	12	11	11	11	10	10	9	9	8											
20	27	24	22	21	20	19	18	17	16	16	15	15	14	14	13	12	12	11	11	11	10	10	9	9	9				
25	33	30	28	26	24	23	22	21	20	19	18	18	18	17	16	15	15	14	14	13	13	12	11	11	11	10	10	9	25
30	40	36	33	31	29	28	26	25	24	23	22	22	21	21	19	18	18	17	17	16	15	15	14	13	13	12	12	11	30
35	47	43	39	36	34	32	30	29	28	27	26	25	25	24	23	22	21	20	19	19	18	17	16	15	15	14	14	13	35
40	54	49	45	42	39	37	35	34	32	31	29	29	28	27	26	25	24	23	22	21	20	19	18	17	17	16	16	15	40
45	61	55	51	47	44	42	40	38	36	35	33	33	32	31	29	28	26	25	25	24	23	22	21	19	19	18	18	17	45
50	67	60	56	52	49	46	44	42	41	39	37	36	35	34	32	31	30	28	28	27	25	24	23	22	21	20	19	19	50
60	80	73	67	62	58	55	53	50	49	47	44	44	42	41	39	37	35	34	33	32	30	29	27	26	26	25	23	22	60
70	93	85	78	73	68	65	62	59	57	55	52	51	49	48	45	43	41	40	39	37	35	34	32	30	30	29	27	26	70
80	107	97	85	83	78	74	70	67	65	62	59	58	56	55	52	49	47	45	44	43	41	39	37	35	34	33	31	30	80
90	120	109	100	93	88	83	79	76	73	70	66	65	63	62	58	55	53	51	50	48	46	44	41	39	38	37	35	34	90
100	133	121	111	104	97	92	88	84	81	78	74	73	70	68	65	63	59	57	55	53	51	48	46	43	43	41	39	37	100
110	147	133	122	114	107	101	97	92	89	86	81	80	77	75	71	68	65	62	61	58	56	53	50	47	47	45	43	41	110
120	160	145	134	125	117	111	106	101	97	94	88	87	84	82	78	74	71	68	66	64	61	58	55	52	51	49	47	45	120
130	174	157	145	135	127	120	114	109	105	101	96	94	91	89	84	80	77	74	72	69	66	63	60	56	55	53	51	48	130
140	191	173	159	148	138	131	124	119	113	109	103	102	98	96	91	86	83	79	77	74	71	68	64	60	60	57	55	52	140
150	205	185	170	158	148	140	133	126	123	117	110	109	104	103	97	92	89	85	83	80	76	73	69	65	64	61	58	56	150
160	218	197	181	168	158	149	142	135	130	125	118	116	113	109	104	99	94	91	88	85	81	77	73	69	68	65	62	60	160
180	245	221	203	189	177	168	159	152	146	140	133	131	127	123	116	111	107	102	99	96	91	87	82	78	77	74	70	67	180
200	273	246	226	210	197	187	177	169	162	156	147	145	141	137	129	123	118	113	110	106	101	97	92	86	85	82	78	75	200

TABLE 7. SETTING OF VALVES CALIBRATED IN SQUARE FEET OF RADIATION—*Continued*

Square Feet of Radiation	Section 2													Height Above Midpoint of Heater in Feet														Section 2	Square Feet of Radiation
	4½–5	5–6	6–7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	18–20	20–22	22–24	24–26	26–28	28–30	30–33	33–36	36–40	40–43	43–46	46–50	50–55	55–60	
10	17	15	14	13	12	12	11	11	10	10	9	9	9																10
15	26	23	21	20	18	17	17	16	15	15	14	14	13	13	12	12	11	11	10	10	9	9	9						15
20	35	31	29	27	25	23	22	21	20	20	19	18	18	17	16	15	15	14	14	13	13	12	11	11	11	10	10	9	20
25	44	39	36	33	31	29	28	27	25	25	24	23	22	21	20	19	19	18	17	16	16	15	14	14	13	13	12	12	25
30	52	47	43	40	37	35	33	32	30	30	28	28	27	26	24	23	22	21	20	20	19	18	17	16	16	15	14	14	30
35	63	56	51	47	44	41	39	38	35	35	33	32	31	30	28	27	26	25	24	23	22	21	20	19	18	18	17	16	35
40	71	64	58	54	51	47	45	43	41	39	38	37	35	34	33	31	29	28	27	26	25	24	23	22	21	20	19	18	40
45	83	73	66	61	57	54	51	49	46	44	43	41	40	39	36	35	33	32	31	30	28	27	26	25	24	23	22	21	45
50	88	79	72	68	62	59	56	53	51	50	48	46	44	43	41	39	38	35	34	33	34	30	28	27	26	25	24	23	50
60	106	95	87	80	75	71	67	64	61	59	57	55	53	52	49	47	44	42	41	39	38	36	34	33	32	30	29	28	60
70	126	112	102	94	88	83	78	75	72	69	66	64	62	60	57	54	52	50	48	46	44	42	40	38	37	35	34	32	70
80	144	128	116	107	100	95	90	86	82	79	76	74	71	69	65	62	59	57	54	53	50	48	45	44	42	40	38	37	80
90	159	142	130	120	112	106	101	96	92	89	85	83	80	77	73	70	66	64	61	59	56	54	51	49	47	45	43	41	90
100	177	158	144	133	125	118	112	106	100	99	95	92	89	86	81	78	74	71	68	66	63	60	57	55	53	50	48	46	100
110	195	174	159	147	138	130	123	117	113	109	105	101	98	95	90	85	81	78	75	73	69	66	62	60	58	56	53	51	110
120	212	190	173	160	150	142	134	128	123	119	114	110	106	103	98	93	89	85	82	79	75	72	68	66	63	61	58	55	120
130	230	206	188	173	163	153	145	138	133	128	123	120	115	112	106	101	96	92	89	85	82	78	74	71	69	66	63	60	130
140	260	230	208	192	179	168	160	152	143	138	133	128	124	120	114	110	104	99	95	92	88	83	79	76	74	71	67	64	140
150	279	246	223	206	192	180	172	163	154	148	142	138	133	129	122	116	111	106	102	99	94	89	85	82	78	76	72	69	150
160	296	262	238	219	214	192	182	173	164	158	152	147	142	138	130	124	118	114	109	105	100	96	91	87	85	81	77	74	160
180	332	294	267	246	230	216	204	195	184	178	171	165	160	155	146	140	133	128	123	119	113	107	102	98	95	91	87	83	180
200									225	197	190	184	178	172	163	155	148	142	136	132	126	119	113	109	106	101	96	92	200

TABLE 7—SETTING OF VALVES CALIBRATED IN SQUARE FEET OF RADIATION—*Continued*

SQUARE FEET OF RADIATION	SECTION 3											HEIGHT ABOVE MIDPOINT OF HEATER IN FEET															SECTION 3		SQUARE FEET OF RADIATION
	4½–5	5–6	6–7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	18–20	20–22	22–24	24–26	26–28	28–30	30–33	33–36	36–40	40–43	43–46	46–50	50–55	55–60	
10	24	21	19	17	16	15	15	14	14	13	13	12	12	11	11	10	10												10
15	35	31	28	24	23	23	22	22	20	20	19	18	18	17	16	15	14	14	13	13	12	12	11	11	10	10	9	9	15
20	49	43	39	36	33	31	29	28	27	26	25	24	24	23	21	20	19	18	18	17	16	16	15	14	14	13	12	12	20
25	61	54	49	45	42	39	37	35	34	33	31	30	29	28	27	25	24	23	22	21	20	19	18	18	17	16	16	15	25
30	73	65	58	54	50	47	44	42	41	39	38	36	35	34	32	30	29	28	27	26	25	23	22	21	20	20	19	18	30
35	90	78	70	64	60	56	53	50	48	46	44	46	41	40	37	35	33	32	30	30	29	27	26	25	24	23	22	21	35
40	106	89	80	73	68	64	60	57	55	52	50	48	47	45	43	41	39	37	30	34	33	31	29	28	27	26	25	24	40
45	121	104	92	84	78	73	69	65	61	59	57	54	53	51	48	46	43	42	40	39	37	35	33	32	31	29	28	27	45
50	125	110	99	91	84	79	74	71	68	65	63	61	59	56	53	51	48	48	45	40	41	39	37	35	34	33	31	30	50
60	150	131	119	109	101	95	89	85	82	78	75	73	70	68	64	61	58	56	53	51	49	47	44	42	41	39	37	36	60
76	181	157	141	128	119	112	105	100	96	91	88	85	82	79	75	75	68	65	62	60	57	55	51	50	48	46	44	42	70
80	207	180	161	147	136	127	120	114	110	105	100	97	94	90	85	81	77	74	71	69	65	62	59	57	55	52	50	48	80
90	227	298	178	164	152	144	134	127	123	118	113	109	105	102	96	91	87	83	80	77	73	70	66	64	61	51	56	54	90
100	252	210	148	184	168	158	149	140	137	131	126	122	117	113	107	101	97	93	84	86	82	78	75	71	68	65	62	60	100
110	277	242	218	199	185	174	164	154	151	144	138	133	129	125	117	112	106	102	98	94	90	86	81	78	75	72	69	65	110
120	210	264	238	218	202	190	179	169	164	157	158	145	141	136	128	122	116	111	107	106	98	94	88	85	82	78	75	71	120
130		286	257	236	219	206	194	183	178	170	164	157	152	147	139	132	126	120	116	112	106	101	96	92	89	85	81	77	130
140			294	267	245	229	216	204	191	183	176	170	164	158	150	142	135	130	125	120	114	109	103	99	96	92	87	83	140
150			315	286	264	245	228	219	205	196	189	182	176	170	160	142	145	139	134	128	122	119	110	107	102	98	94	90	150
160				302	279	260	249	232	219	209	202	194	187	181	171	162	155	148	142	137	131	125	117	114	110	105	100	95	160
180					315	298	271	262	256	235	226	218	211	204	182	173	174	167	168	154	147	140	132	128	123	118	112	107	180
200						326	303	294	274	262	256	242	234	226	214	202	193	185	178	171	163	156	147	142	136	131	125	119	200

TABLE 7—SETTING OF VALVES CALIBRATED IN SQUARE FEET OF RADIATION—*Continued*

Square Feet of Radiation	Section 4					Height Above Midpoint of Heater in Feet																					Section 4		Square Feet of Radiation
	4½–5	5–6	6–7	7–8	8–9	9–10	10–11	11–12	12–13	13–14	14–15	15–16	16–17	17–18	18–20	20–22	22–24	24–26	26–28	28–30	30–33	33–36	36–40	40–43	43–46	46–50	50–55	55–60	
10	34	30	27	24	23	22	21	20	20	19	18	18	17	16	15	14	14	13	13	12	11	11	10	10	10	9	9	8	10
15	52	45	41	37	34	32	31	30	30	29	27	26	25	24	23	22	20	20	19	18	17	16	15	15	14	14	13	12	15
20	77	65	38	52	43	45	42	41	40	38	37	35	34	33	31	29	27	26	25	24	22	22	21	20	19	18	17	16	20
25	95	81	71	64	59	55	52	49	50	48	46	44	42	41	38	36	34	33	31	30	28	27	26	25	24	23	22	21	25
30	130	108	93	83	76	70	66	62	60	57	55	53	51	49	46	43	41	39	38	36	33	33	31	30	29	27	26	25	30
35	156	125	108	97	88	82	77	72	70	67	64	61	59	57	53	50	48	46	44	42	39	38	36	35	33	32	30	29	35
40	200	155	132	116	105	97	90	85	30	77	73	70	67	65	61	58	55	52	50	48	45	43	41	39	38	36	35	33	40
45	226	175	149	131	119	109	101	96	90	86	82	79	76	73	69	65	62	59	56	54	50	49	46	44	43	41	39	37	45
50	220	170	150	135	124	115	107	100	100	96	91	88	84	82	76	72	68	65	63	60	56	54	51	49	47	45	43	41	50
60	266	216	187	167	152	141	131	124	120	115	110	105	101	98	92	86	82	78	75	72	67	65	62	59	57	54	52	49	60
70	310	251	218	195	178	165	153	145	141	134	128	123	118	114	108	101	96	91	88	84	78	76	72	69	66	63	60	57	70
80		325	282	246	220	201	187	174	161	153	146	140	135	130	122	115	110	104	100	96	90	87	82	79	76	73	69	66	80
90			300	234	226	201	194	183	181	172	165	158	152	147	137	129	123	117	113	108	101	98	92	89	86	82	78	74	90
100				301	271	250	230	216	201	191	183	175	169	163	153	144	131	130	125	120	112	108	102	99	95	91	96	83	100
110					300	274	254	234	221	210	206	193	185	180	168	158	150	143	138	132	123	119	113	109	104	100	95	91	110
120						300	278	260	241	230	219	210	202	196	183	173	164	157	150	144	134	130	124	118	114	109	104	99	120
130							300	280	261	249	238	228	219	212	198	187	178	170	162	156	146	141	133	128	123	118	112	107	130
140								301	281	268	256	245	236	228	214	202	191	183	175	167	157	152	144	138	133	127	121	115	140
150									301	287	274	263	253	245	229	216	205	196	187	181	168	163	154	149	142	136	130	124	150
160										303	292	280	270	261	244	230	218	209	200	193	179	174	164	158	152	145	138	132	160
180											329	315	304	294	274	259	246	235	225	217	202	195	185	177	171	163	155	148	180
200															305	288	273	261	250	240	224	217	205	191	190	182	173	165	200

wise be required. An even better arrangement, when it can be used, is to eliminate the bend and run the lateral portions of the risers so that one is directly above the other as illustrated in Fig. 4B. For a system of this size it is probably best to arrange the data as shown in Table 8, on page 275.

The numbers in column 1 of Table 8 give the radiator numbers consecutively beginning at the end of the branch farthest from the heater. The numbers in column 2 give the section of the line in which the radiator is located. The numbers in column 3 represent the height of the mid-point of each radiator

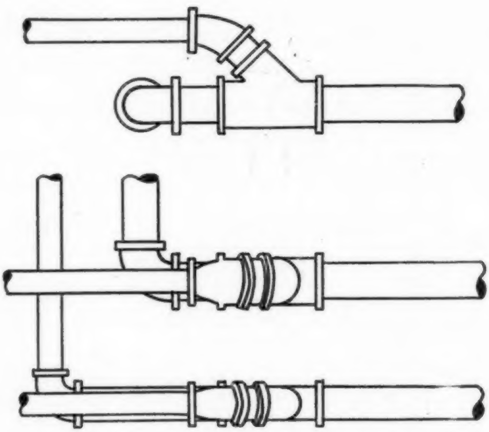

FIG. 8. METHOD OF TAKING A SHORT BRANCH OFF THE TOP OF A LONG BRANCH

above the mid-point of the heater. Thus, Radiator 11 is 8 ft 0 in. above the mid-point of the heater, while Radiator 10 is 7 ft 4 in. above the mid-point of the heater. The heights vary a trifle even among radiators on the same floor because some of the radiators are taller than others. The numbers in column 4 of the tabulation represent the number of square feet of radiation furnished by each radiator. Number 11, for instance, has 160 sq ft, No 10 has 24 sq ft and so on down the column.

The numbers in column 5 represent the number of square feet of radiation that the section of the mains immediately preceding the radiator must carry. Thus, the mains immediately preceding No. 11 must carry 160 sq ft, the section preceding No. 10, 184 sq ft and that preceding No. 9, 249 sq ft. Each number in this column is obtained by adding together the number on its left and the number immediately above it. Thus, 184 is obtained by adding 24 to 160, 249 is obtained by adding 65 to 184, and 270 is obtained by adding 21 to 249. The only exceptions to this are 676 and 981. These represent the number of square feet that must be carried by a section immediately preceding a junction. They are obtained by adding together the numbers of square

feet carried by each branch. Thus, 676 is obtained by adding 208 to 468, and 981 is obtained by adding 230 to 751.

The numbers in columns 6 and 7 are riser sizes. They are obtained by selecting the sizes from Table 1 that correspond to the number of square feet for each radiator as given in column 4. The numbers in columns 8 and 9 are the sizes of the mains for the sections whose capacities are given in column 5. In this system the height from the mid-point of the heater to the mid-point of the lowest portion of the mains is 3 ft 6 in. Therefore, Table 2 is used to determine the sizes of the mains.

The length of a branched line must always be measured along the longest branch; consequently the length of the line in this case is 3 ft + 15 ft + 31½ ft + 4½ ft + 8¼ ft + 10 ft or 72¼ ft. Consequently, the proper number of square feet is indicated in column 3 under the column headed "70-80." Thus, for 160 sq ft, find that a 2½ in. flow main and a 2½ in. return main are required. For 184 sq ft a 2½ in. flow main and a 2½ in. return main are needed and so on. The short line at the bottom of the figure is 6 ft + 24 ft + 3 ft + 7 ft, or 40 ft long. Accordingly, use the column headed "30-40" to find the sizes of its mains.

TABLE 8. ARRANGEMENT OF DATA FOR DESIGN OF HOT WATER SYSTEM

Radiator Number	Section of Line	Height of Mid-Point of Radiator Above Mid-Point of Heater	Square Feet for Radiator	Square Feet for Mains	Flow Riser (Table 1)	Return Riser (Table 1)	Flow Main (Table 2)	Return Main (Table 2)	Main Heater Flow Riser (Table 4)	Main Heater Return Riser (Table 4)	Orifices (Table 5)
1	2	3	4	5	6	7	8	9	10	11	12
11	4	8'-0"	160	160	1½	1½	2½	2½	......	......	3/4
10	4	7'-4"	24	184	¾	¾	2½	2½	......	......	9/32
9	3	8'-0"	65	249	1	1¼	3	3	......	......	13/32
8	3	17'-0"	21	270	½	¾	3	3	......	......	11/64
7	3	17'-0"	50	320	1	1	3	3	......	......	9/32
6	2	17'-0"	40	360	¾	1	3	3	......	......	7/32
5	2	7'-6½"	51	411	1	1	3½	3½	......	......	9/32
4	2	16'-6½"	36	447	¾	1	3½	3½	......	......	13/64
3	2	17'-0"	21	468	½	¾	3½	3½	......	......	5/32
14	3	7'-6½"	93	93	1¼	1¼	2	2	......	......	15/32
13	2	17'-0"	99	192	1¼	1¼	2½	3	......	......	11/32
12	2	7'-6"	16	208	½	¾	2½	3	......	......	11/64
	Junction of Branches										
....	...	....	....	676	......	......	4	4	......	......	......
2	1	17'-0"	50	726	1	1	4	5	......	......	7/32
1	1	8'-4½"	25	751	¾	¾	4	5	......	3½	11/64
16	4	7'-8½"	125	125	1¼	1¼	......	......	......	......	11/16
15	1	7'-6½"	105	230	1¼	1¼	......	......	......	2½	3/8
	Junc	tion......	.....	981	......	......	5	5	4	......	......
No.	3 &	No. 4...	57	......	1	1	......	......	......	......	......

The numbers in column 10 are the sizes of the main heater risers. They are obtained by reference to Table 4 as previously. The numbers in column 11 are orifice sizes. Before these can be selected, the line must be divided into four equal sections. Since the longest branch is 72¼ ft long, the sections will each be 18 ft 1 in. long. This distributes the radiators among the sections as follows:

Section 1 contains radiators 1 and 2.
Section 2 contains radiators 3, 4, 5, 6, 12 and 13.
Section 3 contains radiators 7, 8, 9 and 14.
Section 4 contains radiators 10 and 11.

In the case of the small circuit at the bottom of Fig. 6:

Radiator 15 is in Section 1.
Radiator 16 is in Section 4.

As previously noted, the section for each radiator is recorded in column 2 of Table 8. All radiators on the first floor except No. 1 come within the range of 7-8 ft above the heater; consequently their orifice sizes are found under the heading "7-8" in the proper sections of Table 5. The orifice size for No. 1 is found in Table 5, Section 1, under the column headed "8-9."

If it had been decided to use orifices calibrated for vapor systems or adjustable valves, the sizes or settings would have been as indicated in Table 9, page 279. There are two features of Table 9 that require comment. It will probably be noticed that the valve settings are always equal to or larger than the largest number in the corresponding vapor orifice rating. There are two reasons for this. First, the orifices, as nearly as could be determined by measurement, were correct at the upper range—that is, the 41-50 was actually 50, the 51-60 was actually 60, and so on. Thus, in comparing a valve setting to an orifice rating, attention should be given to the larger number in the orifice rating. Second, an orifice that is a little too small does no particular harm, while one that is too large may, to some extent, allow the riser pressure head to interfere with circulation. The orifice, therefore, is never too large, but may be as much as ten units too small. By taking both of the above considerations into account, it will be noted that the valve setting corresponding to a given orifice will nearly always be equal to or larger than the largest number on the orifice, but that it can never exceed it by more than 10. The second feature that requires comment is the limited range of the valve. None of them, so far as has been ascertained, can be set higher than 200. A certain amount of overloading will not do any harm and it is quite possible that these valves might be used successfully on radiators requiring a setting as high as 300, but on the other hand, the writer would hesitate to use them on a radiator requiring a setting of less than 30. When small settings are used the orifice in the valve becomes a narrow slit, and, therefore, might quite probably become clogged.

However, it is probable that the smaller settings recommended for these valves are too small because of the elongated shape of the orifice. There has as yet been no opportunity to test this experimentally, but since small readjustments can readily be made while the system is in operation, it should be easy to make satisfactory installations using these valves.

Systems for Buildings More than Two Stories High

It will be noticed that in the two-story system just considered all the first-floor radiators were independently connected to the mains. This was not

only convenient, but necessary. When using these tables every radiator less than 12 ft above the mid-point of the heater must be connected to the mains by independent risers. If this is not done the lower radiators will not heat properly. However, for radiators more than 12 ft above the mid-point of the heater, any number may be connected to the same riser, provided the capacity of the riser is not exceeded.

Fig. 11 shows the layout for one branch of a 3-story installation. For convenience the radiators have been numbered 11, 12, 13, 14, 15, on the first floor; 21, 22, 23, 24, 25, on the second floor; and 31, 32, 33, 34, 35, on the third floor. Thus the first digit represents the floor and the second the riser. The line is 97 ft long and therefore each of the four equal sections into which

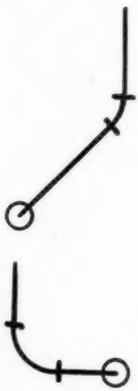

FIG. 9. (*Left*) LOW FRICTION CONNECTIONS FOR RISER LATERAL REQUIRING A BEND

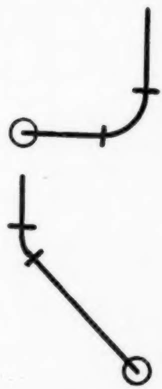

FIG. 10. (*Right*) ALTERNATIVE LOW FRICTION CONNECTIONS FOR RISER LATERAL REQUIRING A BEND

the line must be divided are 24¼ ft long. The distance from the mid-point of the heater to the mid-point of the main directly above is 4 ft 4 in. The calculations are made exactly the same as for the previous cases except that for the second floor radiators the radiator connections and risers must be considered separately. The data for this installation are given in Table 10.

Similar tables can be made for either the vapor rated orifices or adjustable valves. It will probably be noticed that the main risers frequently do not need to be as large as the mains. Except for the increased cost there is no objection to enlarging them. Both the appearance and the circulation will be improved.

HOW TO IMPROVE THE CIRCULATION IN OLD SYSTEMS THAT DO NOT HEAT PROPERLY

It is always desirable, of course, to see the system in actual operation, if possible. It should be carefully checked to make certain that there are no air pockets or other defects that might prevent circulation. Next, by means of Table 2 or Table 3, whichever is applicable, check the capacity of the main to see if it is as great or nearly as great as it should be. If the mains are large enough, and they usually are, the system should be supplied with orifices according to Table 5. If the first-floor radiators have independent risers this

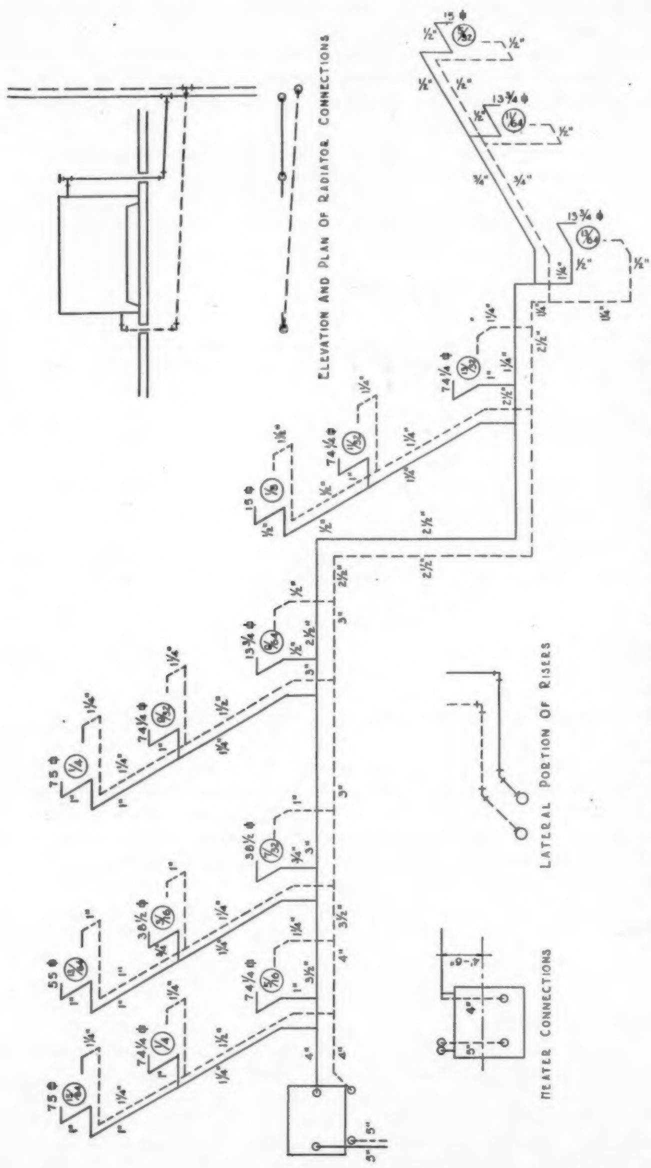

FIG. 11. LAY-OUT FOR ONE BRANCH OF A THREE-STORY INSTALLATION
The sizes of the orifices are indicated by a circle with a number inside

will remedy the trouble. If the first-floor radiators do not have independent risers, it may remedy the trouble anyway, but if it does not, a separate return main for these radiators will make them circulate and may be a cheaper remedy than the separate risers called for by the table. When the mains are too small probably the best solution is to run a separate line to relieve the overloaded mains. The one thing that should always be done is to put an orifice of the size indicated by Table 5 in each radiator.

COVERING THE MAINS

These tables are based on the assumption that the mains are to be bare. Bare mains, particularly bare return mains, help circulation by increasing the heater temperature differential without any corresponding increase in the radiator temperature differentials. This increase in the heater temperature differential causes an increase in its pressure head and thereby helps circulation. It has been found by experiment that better results are usually obtained when the half of the flow main farthest from the heater is covered because the radiators near the end of the line then receive hotter water. For the same reason it has been found desirable to cover the smaller flow risers and flow laterals; ½-in. and ¾-in. flow risers and laterals give much better results when covered. Risers 1 in. or larger are not much affected by cooling, at least in installations of three stories or less. The return main should always be left bare throughout its entire length.

If it is desired to construct a system having an unheated basement making it necessary to cover both pipes, the same tables can be used, provided the line is

TABLE 9—SIZES OF SETTINGS FOR ORIFICES CALIBRATED FOR VAPOR SYSTEMS OR ADJUSTABLE VALVES

Radiator	Section of Line	Height Above Mid-point of Heater	Sq Ft of Radiation	Vapor Rating of Orifice	Valve Setting
11	4	8 ft-0 in.	160	¾ in.	Impossible—must use 2, or better, 3 smaller radiators instead of this large one
10	4	7 ft-4 in.	24	51-60	62
9	3	8 ft-0 in.	65	101-110	118
8	3	17 ft-0 in.	21	11-20	24
7	3	17 ft-0 in.	50	41-50	59
6	2	17 ft-0 in.	40	21-30	35
5	2	7 ft-6½ in.	51	51-60	69
4	2	16 ft-6½ in.	36	21-30	32
3	2	17 ft-0 in.	21	0-10	19
14	3	7 ft-6½ in.	93	151-160	169
13	2	17 ft-0 in.	99	71-80	88
12	2	7 ft-6 in.	16	11-20	21
2	1	17 ft-0 in.	50	21-30	35
	1	8 ft-4½ in.	25	11-20	24
16	4	7 ft-8½ in.	125	1 1/16	Probably would be more satisfactory to use 2 smaller radiators
15	1	7 ft-6½ in.	105	101-110	110

divided into three equal parts instead of four. The fourth section is the one to be eliminated. The effect of this change is to put increased friction into the radiator circuits which tends to prevent them from interfering with each other and thereby enables the reduced pressure head of the heater to keep the water moving through the mains as it should.

Systems Having Sealed Expansion Tanks

Although these tables were designed primarily for systems having open expansion tanks, they will work equally well for systems having sealed expansion tanks. If it is desired to design a system that will run on a lower temperature differential, this can be done by multiplying the number of square feet for each radiator by some constant greater than one, such as 1.5, before beginning to make a tabulation of the data.

Precautions to Be Observed

1. Be sure the mains are large enough, as straight as possible, and installed carefully enough to preclude the possibility of air pockets.

2. Be sure that all radiators less than 12 ft above the midpoint of the heater are connected to the mains by independent risers.

3. Make the lateral portions of the risers as straight as possible.

4. Do not bunch too much radiation in the fourth section of the line. See that the first and second sections get a fair share of the radiation, even if it is necessary to make these two sections a trifle longer than they should be. When a riser comes on the border line between the two sections, it should always be placed in the section nearer the heater.

5. Some contractors make a practice of increasing the size of radiators at the end of the line farthest from the heater in order to compensate for poor circulation. When using these tables that should not be done, because every radiator can be counted upon to deliver about 150 Btu per hour per square foot when the water is leaving the heater at 200 F. If the sizes of some of the radiators are increased, the house will not be uniformly heated.

6. Finally, be sure to put an orifice of the proper size in every radiator. The omission of a single orifice may be sufficient to cause a great deal of trouble.

7. It is a good plan to cover all ½ in. and ¾ in. flow laterals and risers as well as about 1/3 or ½ of the flow main beginning at the end farthest from the heater.

How the Tables Were Calculated

In making the tables a special effort was made to accomplish the following things:

1. To make sure that the radiators at the ends of the lines would get as hot as those near the heater, no matter how long the lines were.

2. To make sure that the radiators on the lower floors would get as hot as the radiators on the upper floors.

3. To provide a safety factor so large that the circulation would be very good, even though the system was somewhat overloaded as, for instance, by adding a radiator or two after the installation had been completed.

4. To make the tables as simple as possible, especially those portions that relate to pipe sizes.

It has been impossible to test the tables experimentally through their entire range, but the tests that have been made have been so very satisfactory that the first three aims have been very largely realized. It is hoped that the fourth will also be accomplished. Attempts to produce simple tables have cost more labor than any other factor.

UNDERLYING CAUSES OF POOR CIRCULATION AND THE REMEDIES ADOPTED

In order to make clear the method of producing strong circulation through the radiators on the ends of the lines, it will be necessary to refer briefly to a paper entitled "Pipe Sizes for Hot Water Heating Systems" by Giesecke and Smith published in the A.S.H.V.E. TRANSACTIONS, 1929. In this paper it was pointed out that the principal cause of cold radiators on the end of a line was the high pressure built up in the return main by the radiators nearer the heater. This fact is so important that to ignore it is to invite almost certain failure in certain parts of the system. Just how it affects the calculation of pipe sizes may be seen from the following considerations.

When making calculations on hot water heating systems it seems to have

TABLE 10—SUMMARY OF DATA FOR ONE BRANCH OF A THREE-STORY INSTALLATION

Radiator	Section of Line	Height of Mid-point of Radiator above Mid-point of Heater	Square Feet for Radiator Connections	Square Feet for Risers Carrying more than one Radiator	Square Feet for Mains	Flow Radiator Connections (Table 1)	Return Radiator Connection (Table 1)	Flow Riser (Table 1)	Return Riser (Table 1)	Flow Main (Table 3)	Return Main (Table 3)	Main Heater Flow Riser	Main Heater Return Riser	Orifices
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15
35	4	25'-10"	15			½	½							5/32
25	4	16'-11"	13¾	28¾		½	½	¾	¾					11/64
15	4	8'-1"	13¾			½	½							13/64
					42½					1¼	1¼			
34	3	25'-10"	15			½	½							1/8
24	3	16'-11"	74¼	89¼		1	1¼	1¼	1¼					11/32
14	3	8'-1"	74¼			1	1¼							13/32
					206					2½	2½			
33	2	25'-10"	75			1	1¼							1/4
23	2	16'-11"	74¼	149½		1	1¼	1¼	1½					9/32
13	2	8'-1"	13¾			½	½							9/64
					369¼					3	3			
32	1	25'-10"	55			1	1							13/64
22	1	16'-11"	38½	93½		¾	1	1¼	1¼					3/16
12	1	8'-1"	38½			¾	1	1¼						7/32
					501¼					3½	3½			
31	1	25'-10"	75			1	1¼							15/64
21	1	16'-11"	74¼	149¼		1	1¼	1¼	1½					1/4
11	1	8'-1"	74¼		724¾	1	1¼			4	4	3½[a]	3½[a]	5/16

a May be replaced by 4 in. risers if so desired.

been customary to assume that the flow through the mains was caused by the difference in weight of two columns of water extending from the middle of the heater to the middle of the radiator. For underfoot distribution systems this particular assumption cannot be justified and has undoubtedly been one of the causes of disappointing performance. In order to make a radiator start in the right direction it is necessary that the pressure in the flow main at the point of junction with the flow riser be higher than the pressure at the corresponding point in the return main. If the pressure in the return main is higher the radiator will start backward. It will also be sluggish and never really hot. It is very important then to keep the pressure in the flow main higher than the pressure in the return main. Fig. 12 shows that the pressure head of the heater does tend to raise the pressure in the flow main. The cool, heavy water in the

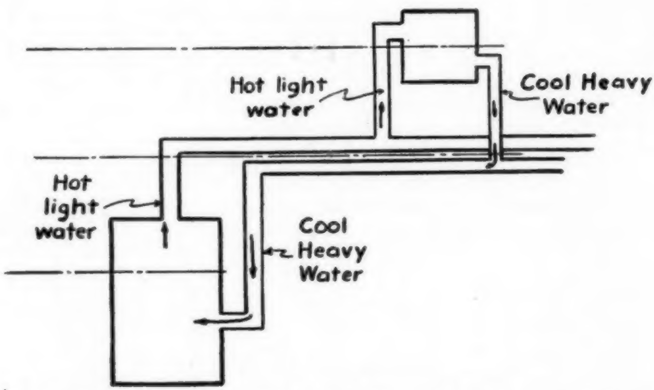

FIG. 12. OPPOSING PRESSURE HEADS OF A HEATER AND RADIATOR CIRCUITS

return heater riser tends to force water into the bottom of the heater and out of the top, thus producing the desired higher pressure in the flow main.

As to the radiator, the cool, heavy water is flowing directly into the return main, thereby tending to increase the pressure in exactly the wrong place. The conclusion is, therefore, that except for its own particular circuit, the pressure head generated in a radiator and its risers, instead of helping circulation, is a great disturbing influence. The problem of insuring good circulation is, then, two-fold.

1. The resistance of the mains must be low enough so that the heater pressure head alone is great enough not only to overcome all the friction in the mains, but also to leave a surplus large enough to insure that throughout the entire length of every line the pressure in the flow main shall be higher than the corresponding pressure in the return main.

2. The friction head in every radiator circuit must be great enough not only to absorb all its own pressure head, but to absorb, in addition a pressure head

equal to the difference in pressure between the mains at the points where its risers join the mains.

Having located the trouble with radiators at the ends of lines, consider some of the difficulties that beset first-floor radiators. In order to make the explanation as clear as possible, note that the line in Fig. 13 has two valves in it, one near the top and the other near the bottom. Under normal conditions the flow riser on the left will be full of light, hot water and the return riser on the right is full of cool, heavy water. Now, if the upper valve is closed, it is evident that the cool, heavy water column in the return riser will displace the column of light, hot water in the flow riser and the radiator will circulate strongly in the right direction. Suppose the upper valve be opened and the lower one closed. The cool, heavy column of water in the return riser now tends to flow backward through the radiator, just the opposite of its former action. From this it is apparent that the two columns of water below a radiator tend to make it flow

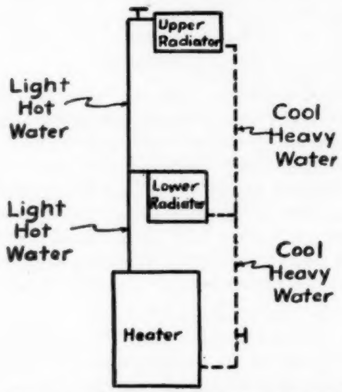

Fig. 13. Opposing Pressure Heads in a Riser

in the right direction, while the columns above it tend to make it flow in the wrong direction. It is evident that if the lower radiator is to circulate properly, the head generated by the water columns above it must be neutralized by adding friction to the upper radiator. Similarly, the friction in the lower pipes should be as small as possible.

Unfortunately, in any actual circuit there are necessarily a number of bends just below the floor. In order to remedy this situation it was decided to choke the higher radiators by means of orifices, but calculations showed that this alone was not sufficient. There were two other things that could be done:

1. Make the lower portions of the risers very large.

2. Avoid placing radiators in the danger zone by connecting all radiators below a certain safe level to the mains by separate risers.

From the standpoint of calculations the first alternative was more simple, but from the standpoint of the tables required it was complicated. Also, it seemed probable that the second method would give a better appearing installation.

It will be noticed that the problem of the low radiator is very similar to the end-of-the-line radiator. In each case a pressure head from some other radiator or radiators is producing interference, and the remedy is two-fold:

1. As far as practicable neutralize the conflicting head by friction.

2. Preserve the strength of the favorable pressure head as far as possible by means of low resistance, or, if this for some reason is undesirable, so arrange the system that no radiators will be connected between points where the pressure is negative.

As far as the mains were concerned, little difficulty was experienced. It was assumed that the heater, its main risers and fittings, including the turns at the tops of the main risers, would absorb 20 per cent of the available pressure head. The capacities given in Table 4 were thus easily selected. It was further assumed that 30 per cent of the heater pressure head was to be available at the ends of the lines. This left 50 per cent to be dissipated in friction in the mains. Accordingly, it was assumed that the mains might have four turns in addition to those already accounted for in the heater connections and the values were calculated for Tables 2 and 3 accordingly.

Since it was considered desirable to leave the mains bare in order to heat the basement, it was inevitable that the water should be somewhat cooled before reaching the farthest radiators. In order to compensate for this it was decided to run the radiators in the different sections of the line at various temperature differentials. The radiators in Section 4 operated on a 25-F differential; those in Section 3 on a 30-F differential; those in Section 2 on a 35-F differential, and those in Section 1 on a 40-F differential. This makes the average temperature of all the radiators about the same and they should all deliver about 154 Btu per hour per square foot of radiation when the water is leaving the heater at 200 F.

The risers all have a friction head of about 16 mill inches per foot, including both pipes. This applies only to the vertical portions. The crooked portions in the basement have a somewhat high and irregularly varying resistance. This makes the pressure to be absorbed by the orifices also an irregular variable so that the capacities cannot be checked by plotting a curve, except for the higher radiators.

It was through the kindness of Prof. C. W. Crawford, head of the Department of Mechanical Engineering and Prof. M. V. Brewer of the same department that it was possible to find a place to install the three story experimental plant. In a recent test on this plant with one-half the flow main and the smaller flow risers covered, the hottest radiator in the system was only 8½ F hotter than the coldest. The average temperature was a trifle over 170 F.

DISCUSSION

ALBERT BUENGER: Several questions that have come to mind are: With a temperature drop of 25 to 40 deg, what is the mean temperature of your radiation? Is the temperature drop figured uniformly for the entire system or do you take a different heat transmission factor for each section of the building?

Just as a matter of information, why do you divide the mains into four sections? For larger houses possibly extending 100 ft from the boiler, do you have more than four divisions?

What about series-connected radiators which are used for an amusement room in the basement on the same level as the boiler? Naturally they will be connected in series with the second floor radiators in order to get proper flow. How are the pipe sizes computed?

E. G. SMITH: An example will answer questions about the mean temperature of the radiators: Suppose that the temperature of the water leaving the heater is 200 deg and it is assumed that the water entered the first section of the radiators at 200 deg and left at 160, a difference of 40 deg and a mean temperature of 180. The second section would change 2½ deg each way, that is the flow temperature would be 197½ deg and the return temperature 162½ deg, at 35-deg difference. The difference in the third section would be 30 deg; the difference in the fourth section 25 deg, although the mean would always be 180 deg. This accounts for the very much smaller orifices used in the first section of the line.

Even though the system were the reverse return type and had the same pressure at all points between the mains you could not get the mean temperature exactly the same for all radiators. If there were 10 or 15 or 20 deg cooling in the mains the temperature of the last radiator would average 20 deg cooler than the first one, an undesirable condition. If one radiator is to run at a 40-deg differential and another farther along the line at 25, it is obvious that considerably more friction must be introduced in the section near the heater. The four sections are partly for that purpose. Also, the pressure on the mains is generated entirely by the heater and is probably three times as much in the nearer section than back farther.

No calculations have been made for radiation in the basement, because it was anticipated that enough heat would be obtained from the mains to meet ordinary basement heating requirements. To meet special cases of basement heating, separate calculations for each system will have to be made by determining the available pressure head in the usual manner and supplying an approximately equal friction head. Knowing the pressure difference required at the tops of the risers at the heater, it is easy to calculate the friction drop to the bases of the risers. The pressure head of the radiator will be the difference between the pressure generated in the parts of the risers that lie above the mains and the parts that lie below the mains. The heights of the risers above the mains must be much greater than the heights of the risers below the mains if satisfactory operation is to be obtained. By putting an orifice of proper size at any place in the line the required friction can be provided.

MR. BUENGER: In other words, you use the system that Professor Giesecke has outlined in his original book?

PROF. SMITH: Yes, that is the only way I know of, to handle the condition described.

The reasons for dividing the mains into four sections are to compensate for cooling and the decrease in pressure toward the end of a line.

H. M. HART: I want to express my gratitude for the progress that has been made on this involved problem of pipe sizes for hot water heating. The

Heating and Piping Contractors National Association have a Standard Committee, whose duty it is to devise standard practices for its members, and there has been an urgent demand for tables of hot water pipe sizes. This committee made several unsuccessful attempts to gather data from which they could compile such tables. Each engineer or contractor has had his own tables, but their origin is obscure.

This is the first time in my experience that a set of tables has been compiled on a really scientific basis. It is very gratifying to see that research engineers can put the results of their studies in such a simple and practical form that it will be usable by those who are not trained engineers. If contractors had engineering training they could use Professor Giesecke's treatise on hot water heating and lay out satisfactory heating systems. The graduate mechanical engineers in my own organization say that Professor Giesecke's rules for designing hot water heating systems are too scientific for ordinary use. That is not a reflection on the work of Professor Giesecke, but rather a tribute to his careful and accurate work.

I have felt that pipe size data in tabular form would be ideal, but doubted that they could be satisfactorily produced. Now I am very optimistic and believe we have started something that will prove very satisfactory and acceptable. More work must be done, but the results so far are splendid.

MR. NICHOLLS: Twenty years ago I earned my living as an engineer; but most of my work has been in contracting, and my comments will be from that viewpoint. As I understand this paper a system of variable sized openings to the branches are used according to the distance from the boiler. Sometimes in a bungalow it is not easy to obtain uniform flow. Do you ever use an overhead system?

PROF. SMITH: That arrangement works very well. However, some heat will be lost in the attic even if the pipes are insulated.

MR. NICHOLLS: The heat loss is immaterial, the great advantage being a uniform flow all over the system whether it is 10 ft or 100 ft away from the boiler. I have put in a great many overhead systems and I never had one that was unsatisfactory. In many cases I know that larger sizes of pipes were used than were necessary. Carpenter's tables were used to the best of our knowledge, but it is less expensive to pay for pipe than to hire an engineer to make absolutely accurate designs and calculations.

No. 861

PANEL WARMING

By L. J. Fowler, London, England
NON-MEMBER

THROUGHOUT travels in over a dozen countries, the author has found that there prevails a popular belief that the Englishman spends his winter evenings in front of a large open fire in a room with fully opened windows. Such a state of affairs, whilst being a libelous exaggeration of the actual conditions in a typical English home, presages an elementary but instinctive conception of the fundamental physiological principles making for the comfort of the human body. In surroundings at 64 F, Rubner states that the mean surface temperature of clothes and exposed skin is about 75 F. Without artificial warming, at a temperature of 60-65F, no sensation of cold is felt, but at a lower temperature than this, radiation and convection are increased and a feeling of chilliness occurs. If, however, the walls and ceiling of a room are maintained at about 75 F, the mean radiation losses from the human body are almost nullified; and although the air temperature may be as low as 50 deg, bodily heat, lost only by convection, is easily balanced by the normal metabolism so that no discomfort is felt. These conditions pertain only to still air, for in draughts convection increases, and rises in temperature of 3 deg and 6 deg are necessary to counteract air movements of 160 and 200 fpm respectively.

To give substantiation to the foregoing statements, it should be remembered that it is possible, although lightly clad, to feel comfortable in a snow field, with an air temperature below freezing, provided that the sun is shining and there is no wind. The heat rays from the sun not only strike the body direct, but are reflected to it from the snow, thus counteracting the direct bodily radiation losses. If, however, the sun is obscured, or direct conduction and convection are increased by air movement, very different sensations are felt.

Reverting to our hypothetical room, as wall temperatures increase, air temperatures can be still lower. If, then, it were possible to reach commercially the ideal of raising the temperature of the whole of the walls to some temperature above 75 F not only would comfort conditions result, but lower air temperatures could be maintained, giving a feeling of freshness, and with it the mental and bodily efficiency and good health of the occupants, who would not suffer from the lassitude inseparable from elevated air temperatures. Furthermore, the cost of fuel for the heating system would be reduced by lessening of the heat lost by changes of air, which, in tall buildings, is very considerable. In

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practice, however, even though the practical difficulties could be overcome, and the walls were evenly warmed to 75 deg, in a short time, although at an initial temperature of 50 deg the air would be warmed 10 deg or more by contact with the walls.

Radiant heaters, such as open fires and luminous electric heaters emit rays of varying intensities which, as well as directly impinging upon the occupants,

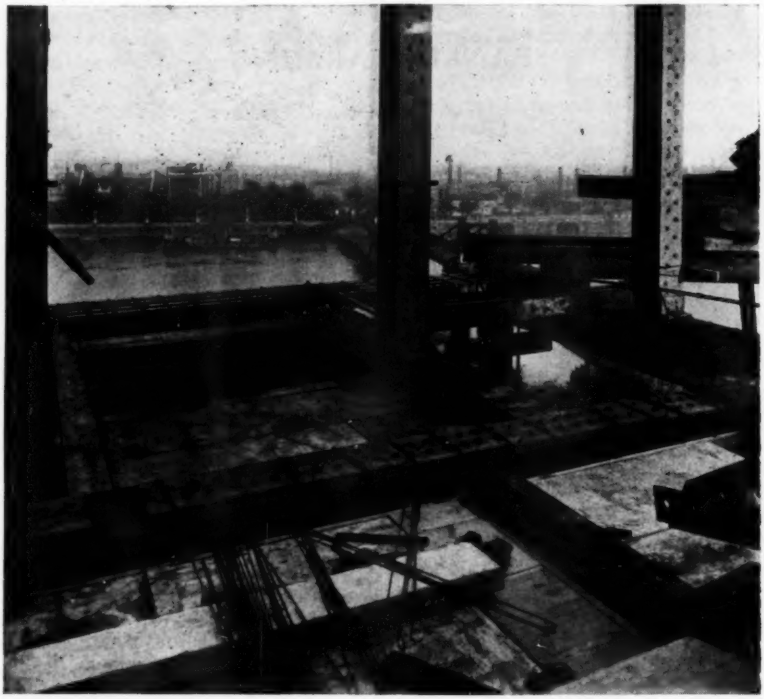

FIG. 1. WARMING PANEL ON SHUTTERING READY FOR CASTING

strike the walls, ceiling, floor and furniture of a room. A portion of this incident energy is absorbed and raises the temperature of the object which it strikes, eventually setting up convection, and a part is reflected one or more times before it reaches any of the occupants. With these forms of heater, it is conceded that, not only is there excessive variation in comfort—and degree of discomfort—throughout the room, but the fuel or power consumption reaches an uneconomic figure. It should be noted at this point that high temperature heat rays have different coefficients of absorbtion and reflection from the longer waves which would be emitted from a low temperature source, such as the walls referred to above, maintained at 75 deg. As an example, as glass is

transparent to light waves, it is largely diathermous to the shorter heat waves; whilst approximately 90 per cent of low temperature heat waves are directly reflected, the remaining 10 per cent partially pass through and partially are absorbed by the glass.

Whilst the higher temperature radiant source is far from satisfactory, the heating of the whole of the walls, ceiling, etc., is not commercially possible, and over 20 years ago Prof. A. H. Barker conceived the idea of the radiator—the word being used in the scientific and not the commercial sense—of comparatively large surface and comparatively low temperature. His patents were taken over and developed by the London firm Richard Crittall & Co., Ltd., to whose efforts is due the very effective compromise called "Panel" Warming, in which large surfaces are warmed to a relatively low temperature which, at times, need not far exceed the 75 deg of the ideal room. On occasion, the walls are warmed, but this is not ideal, for not only is convection set up, but there is limitation to the positions in which heavy pieces of furniture may be placed without obstruction to the uniform distribution of heat. The modern practice is to install the warming surface—termed *panels*—in the ceilings, and this is not only the logical position from all theoretical considerations, but it engenders simplicity and ease of installation as well as other advantages which will be explained later.

The panels generally consist of coils of pipe through which water flows, and the inevitable development of the principle to electrical power has been eminently successful. The pipes, usually of ½ or ¾ in. diameter, run parallel to each other at 4 to 6 in. centres and are embedded in concrete in the underside of the ceiling of the room to be heated. The surface of the concrete is then plastered, rendering the panel invisible. Heat is conducted from the pipes to the plaster face and is then emitted in radiant form. The surface temperature of the plaster is considerably lower than that of the water flowing through the pipes, and is affected by their diameter and spacing. At the centres given, it is found that over the whole of the panel, the temperature varies little.

Allen and Griffith and Davis have found that, with normal plastered and painted surfaces at low temperatures the emissivity is usually within 85-90 per cent of theoretical black body emissivity.

Practice shows, however, that the actual heat lost from warming panels is considerably in excess of that expected from theoretical calculations. There are several reasons for this in a good installation, the most important being air movement. With a ceiling panel fixed flush with the remainder of the surface of the ceiling, in still conditions convection is zero, but should the panel surface be lower than that of the surrounding ceiling, as sometimes occur when an installation has been made in an existing building, or should the material of the ceiling be cracked or porous, a certain amount of convection occurs which slightly increases the total heat emission. If, however, a definite air current blows across the panel, very appreciable increases occur in the total heat emitted, with reduction of the proportion radiated.

Another factor arising to explain the discrepancy between the theoretical and the practical emissions is the heat conduction upwards from the panel. In a normal solid concrete floor construction, in which the concrete is 8 in. thick, the difference in the conductivity between approximately 7 in. concrete above and approximately ⅝ in. plaster below the panel is sufficient, however, to reduce

the upward transmission to a small proportion of the heat lost from the panel. A further reduction is at times effected by interposition above the panel of layers of one of the insulating compounds on the market. A hollow tile floor construction permits less upward loss of heat than a solid concrete, and if the panels are installed in a suspended ceiling, cork or silicate of cotton prove an effective means of retarding the upward flow, and at the same time reduce the deleterious effects of air currents.

In 1909 the first panel warming system was developed, in which the heat was distributed and evenly radiated through a building from within the ceilings, walls and floors. To form radiators which would diffuse heat invisibly, the coils of pipes were embedded in the walls and floors, and although many difficulties had to be overcome, the system was an immediate success. The Royal Liver Building in Liverpool, which has over 1,000 rooms, was the first large building to be equipped in this way, and the system attracted so much attention that a second installation was made in the Midland Adelphi Hotel which was soon erected in the same town. The healthy and pleasant conditions in these two buildings favorably attracted the attention of the medical profession, and panel warming installations were adopted for hospital work.

The war caused a temporary setback, but the boom in building which subsequently began stimulated development of the system, and now it is widely adopted in the best class construction work in England, which, during the past year, includes both the new Bank of England, the Northern Ireland Parliament Buildings, and the London Offices for the Imperial Chemical Industries—the classic of modern British commercial buildings.

The panel system, as generally applied, is a low pressure hot water system, the circulation being either accelerated or by gravity in smaller installations. The distributing mains are designed in a similar fashion to those of a radiator system, but are connected to the panels instead of radiators.

The panels, in the form of continuous coils, are generally fixed in the ceilings, and are of special quality steel pipe. When a reinforced concrete floor is used, the coils are laid on the shuttering, or forms, and connected to the mains which are usually run adjacent to the stanchions. Before delivery to the site, a works test is made at 500-lb per square in. with compressed air under water, but when the coils are connected to the mains, a hydraulic test of 250 lb is applied for several hours before the concrete is cast.

If the floor is constructed by a method which does not employ forms, the coils can be fixed from below, a similar procedure being applied when the system is applied to existing buildings or to false ceilings.

The coils themselves are bent and welded electrically at the works, but all joints during erection are made by the oxy-acetylene welding process.

The heating system of the Royal Liver Building, referred to above, has 55,000 oxy-acetylene welded joints and has stood the test of twenty years.

The heating contracts which were carried out simultaneously by one firm in London three years ago include 75 miles of pipe necessitating 15,850 electric and 13,000 oxy-acetylene welds, all of which have been buried in concrete.

Work on a panel installation commences earlier than that on a similar radiator installation, and close cooperation between the main heating and floor contractors is essential. The fitters follow closely behind the shuttering gang,

laying the panels and fitting the necessary connections to the mains as soon as a bay is shuttered. The welders who follow, join up the connections and then come the testers who mark each bay as it is passed, ready for the concrete of the floor to be poured.

It will be seen that by these means, providing an effort is made to complete the boiler house work and to run the mains, the panel warming system can be set in operation very early, sometimes even before the walls are constructed.

The value of a heating system at so early a stage is immensurate, for not only are working conditions improved for the operatives of other trades, but drying out of concrete and plaster ready for the reception of joinery, etc., allows the progress of the whole construction to be advanced at an abnormal rate. In some cases, it is even profitable to install temporary boilers and mains to obtain this early heat.

In most reinforced concrete or hollow tile constructions, the coils are cast in concrete in the soffit of the floor over that which has to be heated. The temperature of the water gravitating or pumped through the system is not sufficiently high to set up expansion stresses which cannot be taken up by concrete without movement or cracking.

When the concrete is sufficiently matured, the ceilings are hacked if no satisfactory constructional key has been provided, and plaster is applied according to a specification set out by the patentees of the system, as a precautionary measure against cracking which takes place if the materials or workmanship are of inferior quality. If reasonable care is taken, the failure of plaster work in contact with panel surfaces maintained at the normal working temperature need not be feared, and reputable plastering firms in England will give a five-year guarantee for work of this type.

Application of panel warming is not confined to concrete construction. In Devonshire House, an apartment building erected in London some three years ago to the design of Messrs. Carrere & Hastings of New York City, in conjunction with Professor Reilly of Liverpool, the panels heating the three top floors were incorporated in Hy-rib suspended from the constructional floors. In cases such as this, the coils are fixed over and wired to the Hy-rib; plaster is then applied, care being taken to force this through the mesh to obtain an intimate contact with the pipes to allow free conduction of heat from the latter to the plaster face. In order to reduce convection currents and radiation to above, silicate of cotton is carefully laid above the panels to a depth of about 6 in.

Standard arrangements have been developed for incorporation of coils behind marble walls, floors or ceilings behind terrazzo walls and floors, below wood joist floors and finally to existing ceilings, by means of special pre-cast plaster slabs which contain the coils, and which are finished in conformity with the decorations of the rooms in which they are fixed.

Other types of panel can be applied to surfaces of walls and ceilings of new or existing buildings. The pipes are attached to either cast iron or steel plates; in the former case the pipes are cast with the plates in standard sizes, which are connected together with right and left hand nipples. In these Rayrads as they are called, the heat from the pipes is conducted through the plate, the variation in temperature over the whole plate being between 10 deg and

20 F. Excellent results are obtained, and the surface of the plates can be treated in numerous ways to match decorations. The standard sizes of the castings, however, frequently present difficulties in application. For surface application coils similar to those normally cast in concrete are sometimes attached by rivetted bands or by spot welding to rolled sheet steel plates, to give as

FIG. 2. WARMING PANELS APPLIED TO A HY-RIB CONSTRUCTION

intimate a thermal contact as possible. These panels can, of course, be made of any size and present less difficulty in connecting up than Ray-rads. Both the Ray-rad and sheet steel panels can be used with steam, hot water, or water heated under pressure to above the normal boiling point, and thus offer a wide field of application in commercial buildings, factories, etc.

The consumption of electrical power in England has doubled since 1922, and upon the completion of the Government development scheme in which the power stations are centralized into 150 inter-linked units, an even greater

stimulus will be given to an application of Panel Warming which is finding favor in an ever increasing field.

Electric panels of the Dulrae type are formed of a light asbestos sheet faced with cork, on which is placed a layer of plaster composition about ¼ in. thick, containing a grid work of specially insulated wire element, making a panel of a total thickness of about 1½ in. The panels are made in stock sizes, 36x12 in. and 72x24 in. and for various voltages up to 250; they can be operated by either direct or alternating supplies. In existing structures, a number of standard units, connected to form a complete panel of suitable dimensions, are finished with a neat beading, and screwed to the surface of ceilings or walls. In a new structure or one where cutting away does not involve serious difficulties, the panels can be sunk flush with the finished ceiling or wall face. At times, the elements are fixed behind mirrors, in portable folding screens and, if special precautions are taken for waterproofing, in floors of bathrooms. The power absorption is calculated to give a surface temperature of about 120 deg to 130 F on continuous running, but when thermostatically controlled, the mean temperature is in the neighborhood of 80 to 90 F.

Dulrae panels are either switched on independently or arranged for series parallel control in the ordinary way, giving 100 per cent, 50 per cent or 25 per cent of the maximum heat available.

The most efficient method of control is by means of a thermostat operating direct or alternatively through a contactor switch when the load is great.

A compromise between the hot water and the Dulrae Panel is gaining popularity; heating elements are encased in sheaths of glass which are placed inside steel tubes, the space between the two being filled in with a plaster material to permit ready conduction from one to another. The tubes are cast into ceilings or walls, or at times into floors; the positions that they occupy and the results obtained from them are similar to those of hot water panels.

Another form of electric panel, called the Morganite Panel, has useful spheres of application, but although not luminous, the working temperature is higher and the human body is sensitive to the heat rays emitted.

Electric heating especially lends itself to measurement of power and from installations of this type much has been learnt of the fuel consumption of panel systems.

By the courtesy of J. L. Musgrave and R. G. Crittall, to whose foresight and ingenuity, and above all, to whose fortitude, the development of the panel warming is mainly due, I am enabled to give below some figures obtained on an electric panel installation in London during the severe winter of last year. The heating season was prolonged from October 1 to May 1, a period of 212 days, with an average recorded temperature of 43 F. The building under test contained eight large rooms, hall, staircase, etc., and by thermostatic control was maintained at 60 F at which temperature the radiant heat secured conditions of comfort. The total connected load of the electric panels was 20 kw, making 101,760 kwh available throughout the 212 days. The actual power consumed was 30,300 units, approximately one-third full load. The cube of the rooms warmed was 21,250 cu ft, making a total of 1.42 units per foot cube—a moderate annual charge.

FIG. 3. WARMING PANELS APPLIED TO A BARREL CEILING

A northerly room of 3,250 ft cube and equipped with 65 sq ft of heating surface gave an average daily consumption of 10 units for 24 hours, costing approximately one-third as much as a gas fire in an adjoining building.

Experience proved that panel warmed buildings are run with the minimum fuel consumption when they are maintained at a steady heat temperature throughout the winter. Serious losses of fuel or electric power result from rapid heating after prolonged cooling, but if ample radiating surface is installed, the maintenance of a steady and moderate temperature, besides being most economic, gives the greatest comfort to the occupants.

In England, it is found that under these circumstances the hourly consumption is about one-third of that of an intermittent service—a readily computable saving for a building in occupation for twelve hours a day.

Owing to the almost complete absence of convection, hot strata of air are not formed in the upper portions of rooms warmed by ceiling panels, and there is thus a palpable fuel economy, for less heat is lost from convection and overwarmed air escaping through the upper portions of windows.

The prime cost of a low pressure hot water panel installation exceeds by only a small margin that of a similar radiator system if the expense of enclosing and decorating the exposed portions of the radiator system is considered. Furthermore, the value to other trades of the early provision of heat to the unfinished building cannot be readily estimated, but there is no doubt that in modern construction, in which speed is a paramount consideration, it definitely justifies increased expenditure on the heating system.

Architecturally, the system is supreme. The complete invisibility of the radiating surface and of the risers is only relieved by the control valves, fixed in suitable positions within metal access boxes, and by means of which the panels can be controlled from the rooms which they warm. No space is wasted and no unsightly marks of dust arising from convection despoil the decorations of the walls and ceilings. It is frequently desirable in modern commercial buildings, especially those which are sublet as offices, to modify the positions of partitions, a condition to which panel warming is readily adaptable, for the heating surface is uniformly distributed and each bay with its requisite heating surface and control valve or valves is an independent unit.

The architectural virtue of invisibility is claimed by some to engender a vicious element. Unless the greatest care is taken in the preparation of accurately detailed record plans, difficulty may subsequently be experienced in locating the exact positions of the pipes, and alterations to the system are therefore difficult. These, however, are rarely necessary, for the reason given above. In a well designed installation, sub-division by partitions is relatively unimportant.

There are two matters to which reference should be made at this point, namely, expansion and corrosion. The coefficients of expansion of concrete and mild steel are 0.62×10^{-5} and 0.66×10^{-5} per degree Fahrenheit respectively, and the stresses set up in the concrete owing to the higher expansion coefficient of the steel, at the normal working temperature of 120 deg are not sufficient in magnitude to cause failure or cracking.

Internal corrosion is not serious in a system where the make-up water is not great. Oxygen is slowly admitted through valves and the pumps in an accelerated circulation, but experience does not show that the quantities are sufficient

Fig. 4. Warming Panels Ready for Casting Into Walls

to be dangerous. The causes of external corrosion have to be studied by the steelwork and reinforced concrete contractors as well as the heating engineers, and no trouble is experienced if the materials in which the pipes are encased are selected with care. It is important, however, to avoid any oxy-chloride compounds.

Reference has already been made to the economy of maintaining a steady supply of heat. A panel warmed building shows remarkably steady temperatures, in spite of severe external conditions. The building fabric which is warmed throughout, has a large heat capacity; the reaction, therefore, to any sudden drop in temperature is slow, and if control is effected by external thermostats, increase of flow temperature compensates the additional heat loss without affecting the comfort of the occupants.

On behalf of the Industrial Fatigue Research Board, Dr. Vernon has carried out a series of tests in Panel warmed rooms during the last three years. As well as with thermometer readings, observations were made with Moll's Thermopile and the Kata thermometer. Remarkable evenness of conditions existed throughout a room warmed with a ceiling panel fixed near to the window for there was no more than 1 deg of difference of temperature between the floor and the ceiling. Estimation of wall temperatures obtained with the thermopile indicated that the mean wall temperature was from 2 deg to 3 deg higher than that of the air. Subjective observations made during the same series of tests showed that a temperature of comfort was only obtained in a panel warmed room at 1 deg lower than in a room heated by convection.

Actual experiments upon the heat losses from people and the practical experience of the installers of the system indicate that Dr. Vernon's figure for the permissible lowering of air temperature is underestimated.

In the last few years, remarkably even temperatures have been maintained in panel warmed buildings in which the boilers have only been fired for about one-third of the day.

During this period, the excess heat generated is circulated to large storage tanks, where the temperature of the water rises as high as 280 F under the head of the building. An automatic mixing valve, controlled by a thermostat fixed in the flow pipe, permits the water circulating through the system to be continuously maintained at the required temperature, generally between 90 and 120 F by mixing with it a proportion of water from the storage tanks, which gradually fall in temperature until the boilers are once more operated. In some installations, the storage tanks themselves are directly heated by thermostatically controlled immersion heaters which are operated on the night load by means of time switches.

Satisfactory results are also resulting from the thermostatic control of individual rooms by solenoid operated valves controlling the flow of water through the panels. From economic reasons, a low pressure, generally either 8 or 25 volts, is chosen for the electrical supply to the valves. In small installations, batteries are generally used, being fed from trickle chargers, but in larger installations, rotary converters are more satisfactory. The thermostatic element is a bimetallic strip, which makes the circuit and thus closes the valve when the room is warmed to the temperature indicated on the indicator dial of the thermostat, which is set by hand. The circuit is broken when the temperature falls 1 to 1½ F and the valve opens. Where one thermostat controls more than

four valves, it is usual to employ a relay, since the lag between the opening and closing temperatures is otherwise increased. Continuous recording thermometers placed in the rooms controlled in this way show variations of internal temperature of less than 2 deg throughout 24 hours, although the external conditions have changed abnormally. A sufficient testimonial to the efficacy of the thermostatically controlled solenoid valve in conjunction with Panel Warming is the contemporary installation of this dual system in a commercial building in London containing approximately 1,700 rooms.

A further development of fundamental importance is the use of panel systems for cooling during summer months. Cooling water at a temperature of between 40 and 50 F is circulated through the pipes and only a very slight lowering of room temperature is required. The cooled panel surface absorbs the heat radiated from occupants, walls, floors, etc., and there is a definite absorption from convective currents which at the same time prevent condensation at the working temperatures found desirable.

It would be invidious for me to describe the panel system to you as the panacea for all troubles of the heating engineer. Mention has already been made regarding the inaccessibility of the installed system, and careful planning and erection are inseparable requirements of its successful application. Present practice is the result of 20 years of continuous experiment, during which many difficulties have been successfully overcome. In every installation, careful supervision is necessary to ensure that the coils are levelled to prevent air locks and hydraulically tested before they are connected or otherwise made inaccessible. Furthermore, external corrosion must be considered in the selection of materials in which the pipes are encased. In the Bank of England, which is now being rebuilt, every effort is being made to ensure longevity, and it is an interesting fact that a panel system of copper pipes has been chosen.

Supervision of the plastering is desirable to ensure that the correct mixtures are used and the work is well executed. It is the general practice to work a scrim into the finished face of the plaster surface to the panels, and it is essential that care is taken that slovenly workmen do not apply a thin plaster coat after the scrim, as the continuity of the setting coat is then broken, resulting in a tendency towards the formation of cracks. Furthermore, if the heat has already been in operation, it must be shut off several days before plastering and not turned on again until at least four days after the application of the setting coat, the valves being cracked slightly and gradually opened day by day. On large contracts, it is usual to detail a man to keep in close touch with the plaster work, to see that the panels are turned off when necessary and only opened slowly. At the same time, it is his duty to take a note of the materials and application of the plaster itself.

The boom in building and the rapidly enhanced popularity of panel warming after the war at first caused an insufficiency of oxy-acetylene welders. To meet the growing demand, instruction in this comparatively new trade was undertaken both by the larger contractors and at schools, and now there is constant work for the trained man.

Standardization of coils and connections from coils to risers is an essential factor of low costs.

Since the cost of electric welding and bending at the factory is only a small fraction of that of oxy-acetylene welding and bending on the site, the pipes are

there cut to length and bent by suitable dies, and little is necessary on the building itself beyond the welding up in position.

Intensive organization and coordination with other trades bring about ease and rapidity of installation which amply repay the consequent increase of supervision charges. The installation is made continuous and is practically completed before the building construction reaches the difficult final stages. It is then only necessary to maintain a small staff to carry out the testing and regulation and to attend to the running of the plant before the permanent staff takes charge.

Early completion is a factor beneficial not only to the contractor, but to the general progress of the construction as a whole and is fully appreciated by architects.

No mention had yet been made of the development of Panel Warming outside the land of its origination. European extension is progressing rapidly. The licensors have granted agencies in Holland, France, and negotiations are in hand with firms in Germany and Czecho-Slovakia, and Imperial extensions are rapidly increasing. Large contracts have been carried out in New Delhi and elsewhere in India, and contracts are in hand in China and Australia. In this country, as you know, the British Embassy at Washington has just been completed under the auspices of the most famous of modern British architects.

In conclusion, the panel system is a step towards the ideal; at the moment, a somewhat laborious attempt, but a well-founded attempt to apply the methods of nature to the needs of civilized mankind, and twenty years of experience have each brought their improvements which made the difficulties of the previous years worth overcoming.

One instinctively points to the extension of developments in the electrical application of panel warming, but it is impossible to see far, and progress can only be developed step by step.

The principles are those used by the Romans 2,000 years ago with their underground flues, and although methods have changed, it can be forecast with no lack of confidence that future developments will lead to the provision of uniform distribution of radiant heat energy in the successful warming of buildings.

DISCUSSION

F. I. RAYMOND: I have gone over your paper quite carefully and there is one point that you made which was brought out very definitely in your paper, that you only have 1 deg difference between the ceiling temperature and the floor temperature. That is in one installation, I believe. I wonder if you have any theory as to why you do not get a higher temperature at the ceiling than you do at the floor?

R. S. FRANKLIN: I would like to know what method of calculation might be used to determine the proper amount of coil surface? Is it determined in the manner that we ordinarily figure out heat losses? I would also like to know if it is possible to obtain any impression on this side of the water regarding such installations. It is a very interesting subject and everybody who has heard of panel warming has always said that it is only applicable to countries

with mild temperatures. I would like to have Mr. Fowler's thoughts on that subject. If it were applied to our forms of construction here I can visualize cases where it would be impracticable to put on a water test during our winter construction period.

F. M. TORRENCE: A very short question is insulation placed between the coils when they are put in the outside walls.

MR. BRUCE: I was wondering whether the loss in the circulation through the piping, due to the return bend construction, would make it necessary to use a pump running under very high head, or whether the manifold system, as one slide indicated, would be better.

MR. SAUNDERS: I would like to inquire if there is any noticeable difference in the temperature required in heating by radiation from our usual method of heating. That is, we would naturally expect that the room temperature could be lower for the same degree of comfort if the room was heated by direct radiation rather than by convection.

W. H. CARRIER: I would like to make a remark about cooling, referred to in certain climates. I expect that the radiating effect of colder surface, where the dew point is not particularly high, might be quite effective, as pointed out, but certainly where, as we often have even in New York, dew points of 75 deg or even higher, with temperatures of only 80 to 85, on extremely uncomfortable days, we could not get much cooling effect from our walls without getting condensation upon them. I doubt very much whether such application would work. Our temperature of comfort, as pointed out by the speaker, is the result of air temperature producing convection and radiation from surrounding surfaces. If our air temperature is not greatly cooled, we would have to have a correspondingly lower surface temperature. I do not know just what the proportion is, but perhaps something in the ratio of 40 to 60, one way or the other, in order to get the effect we wish.

Comfort temperature is also affected by humidity. If we do not change our humidity we cannot get comfort anyway. Those are conditions that are practical conditions in the U. S. I think they would be still more severe in India under certain conditions, while under other conditions it would work in dry weather. So that as a general proposition it does not look very inviting for a field of investigation.

MR. FOWLER: Mr. Raymond mentioned the case of there being only 1 deg difference in temperature between the ceiling and the floor. If the air in the room is still, that immediately against the panel gets warm and as air is a bad conductor it takes a very long while for that warmth to creep down. The figure given is the result of Dr. Vernon's experiments in a room with still air, in which temperatures were registered by recorders. If you get convection, air movement owing to open windows, or the surface of the panel is below the surface of the ceiling, then greater temperature differences occur between floor and ceiling.

MR. RAYMOND: I do not want you to think that I was questioning the result you gave. It was just simply that it seemed to me there would be more of a tendency for that air to be heated up at the ceiling and I was wonder-

ing whether you had any theoretical explanation as to how the heated air got down into the room.

MR. FOWLER: The heat gets down from the panel to the room largely by radiation. The radiant heat hits the walls and the floor and warms them. If it hits the floor, then you get convection. Your temperature gets remarkably constant throughout the room except for a layer 2 or 3 in. thick underneath the actual panel.

Mr. Franklin on the method of calculation—the same calculations of heat losses are used except that a reduced air change is allowed. Standard heat loss tables which we employ for radiator installations are adopted and the pipes are sized in an exactly similar way.

I think I might answer another question, the question of friction head. The drop in head on a pump job through a panel is somewhere between 3 and 4 ft of water. The jobs are usually sized 20 to 25 ft friction head.

Regarding the application of the system to countries in which the temperatures are very low, we have had no experience beyond this Washington job in this country. In England last year we got temperatures down to 18 F, which means for maintaining internal conditions of 68, we had a 50-deg temperature difference. You calculate heating installations in the Eastern United States, I believe, on a basic temperature of zero degrees, which means a difference of 70. With a typical construction in England, in a room 20 ft square and 10 ft high, with one outside wall and two normal size windows, the normal size for the panel would be about 40 sq ft. That is on a 30-deg difference temperature. If you work on a 50-deg difference, you have 56 sq ft and on a 60-deg difference, 92, and 70-deg difference, 105 sq ft. That would be just over a quarter of the ceiling. I see no reason why the heating should not be effective at 20 deg lower temperature than that which we get sometimes in England.

For testing on the job in cold weather we employ a mixture containing a certain percentage of glycerine and we test one riser at a time. Usually two floors are put in at one time and the mixture is drained out and used over and over again and carried from one job to the other. I doubt whether that plan would be effective in zero temperature.

Regarding coils in the outside walls, we get the same conditions when we put the coils in the ceiling of the top floor. It is usual to install some board form insulation, or a similar compound, and for the top floors we put in a percentage of extra surface to allow for the loss through the roof.

Regarding the difference in the comfort conditions in a room heated by a radiator and a room heated by panels, Dr. Vernon has made some tests and gives a figure just over 1 deg, but general experience proves that it is a matter of 3 or 4 deg, certainly more than Dr. Vernon's figure.

The cooling tests discussed by Mr. Carrier were made during the past summer. We get high humidities in London, but under the highest humidities we could cool down more than 2 deg without condensation. If we tried to cool down 5 deg, condensation occurred. The convection currents retard the formation of drops on the panels and we have no complaint from that. I do not know how our figures would compare with the figures here of 75 deg dew

point, but I do not think that with the very small temperature drop required there would be condensation.

MR. CARRIER: In London you do not get as high humidity as we do here so that your dew points do not run as high as they do here.

MR. FOWLER: I must leave it at that.

Since this paper was prepared, however, panel warming installations have been made to cope with exterior temperatures of 0F, in Germany and Czecho-Slovakia, and others have been installed in France and Italy, in places where the continental climate necessitates guarding against much lower temperatures than are usually experienced in the British Isles.

No. 862

DEVELOPMENT OF A METHOD FOR HEAT REGULATION

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THE purpose of this paper is to describe and explain an apparatus for and a method of automatically controlling the temperature in steam heated buildings. Automatic control should provide a comfortable temperature in all parts of the building at all times, and should reduce operating costs. The apparatus to be described in this paper does this, *first,* by providing a heat control mechanism and, *second,* by providing a method of operation which makes possible an even distribution of steam at reduced heat capacities.

The design of this apparatus is based on fundamentals of heat control going beyond any of the common methods now in use. For this reason its design will be reviewed before describing the apparatus and its operation. The scope of this paper makes it impossible to go as far into the theoretical considerations of temperature control as would be desirable. Many of the statements made would have to be qualified to fit all the situations which might be encountered. These cases can best be handled in discussion.

To maintain a comfortable temperature in a building the considerations are:

1. Heat emission proportional to heat loss.
2. Even distribution of heat.
3. Avoidance of *overshooting* and *lag.*

It should be the purpose of an adequate temperature control apparatus to accomplish these results, but in order to design such an apparatus, the basic means of accomplishing each of these functions must be ascertained. The ruggedness of the apparatus and simplicity of operation are additional factors to be taken into consideration in the design.

The room temperature varies directly as the heat emission of the radiators and inversely as the heat loss of the room. For practical purposes the heat emission of a radiator standing in air at 70 F varies directly with the radiator temperature, and the heat loss from a room at 70 F varies inversely with the outdoor temperature. These relations are not strictly true as the increments of heat emission for each degree of temperature difference between the radiator and surrounding air increase as the temperature difference increases. But since

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this is also true of the heat losses of the room the two errors tend to offset each other.

There are other considerations besides outdoor temperature which affect the heat loss, such as wind and sun. However, this is one case in which the human element, usually far from helpful, actually aids in the solution of the problem. On the windward side of the building the occupants will keep the windows closed, relying on infiltration for ventilation. On the leeward side, especially if it is in the sun, the occupants are apt to have the windows open for ventilation. In this way the factors of wind and sun are somewhat minimized and an outdoor thermometer, protected from the sun, gives a fairly accurate indication of the heat loss. From this it is apparent that the room temperature can

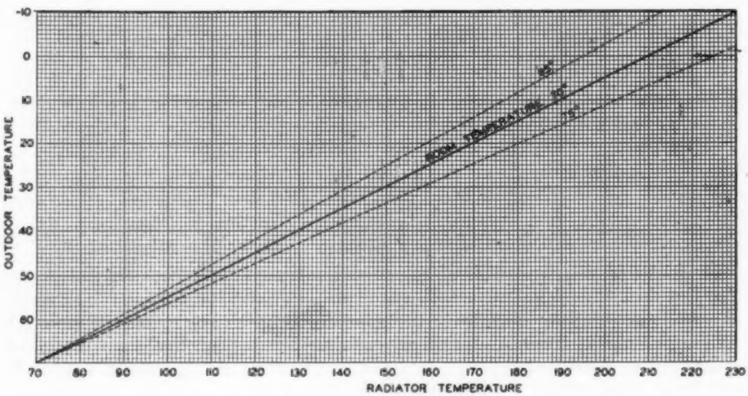

FIG. 1. CURVES SHOWING RELATION BETWEEN OUTDOOR TEMPERATURE, ROOM TEMPERATURE AND RADIATOR TEMPERATURE

be considered to be a function of the radiator temperature and the outdoor temperature. This relation is shown in Fig. 1.

The control of heat emission from the radiator can be effected by (1) varying the steam pressure, (2) fractionally filling the radiator or (3) combinations of these two methods. However, the average temperature of the radiator metal is the only true indicator of the heat which will be emitted by a radiator standing in a room having a temperature of 70 F. For method (1) a single thermometer in intimate contact with the radiator surface will be sufficient to indicate the heat emission at various steam pressures, but this method is limited to a temperature range of about 250 F to 130 F (15 lb pressure to 25 in. vacuum). For methods (2) and (3), two or more thermometers are necessary to give an average temperature of the full and empty portions of a fractionally filled radiator. (See Fig. 2.)

Since method (1) *i. e.,* varying the steam pressure, has the limitation of a 250 F to 130 F radiator temperature, it is necessary that an adequate temperature regulation system utilize fractionally filled radiators to maintain the cor-

rect relationship between radiator temperature and outdoor temperature shown in Fig. 1. A radiator temperature of 130 F is equivalent to a 40 F outdoor temperature. During the greater part of the heating season the outdoor temperature in many localities is above 40 F and a regulation system relying only on vacuum to secure the correct radiator temperature would be ineffectual during this part.

The correct relation between heat loss and heat emission in any one room is of little value unless this relationship is maintained in all parts of the building.

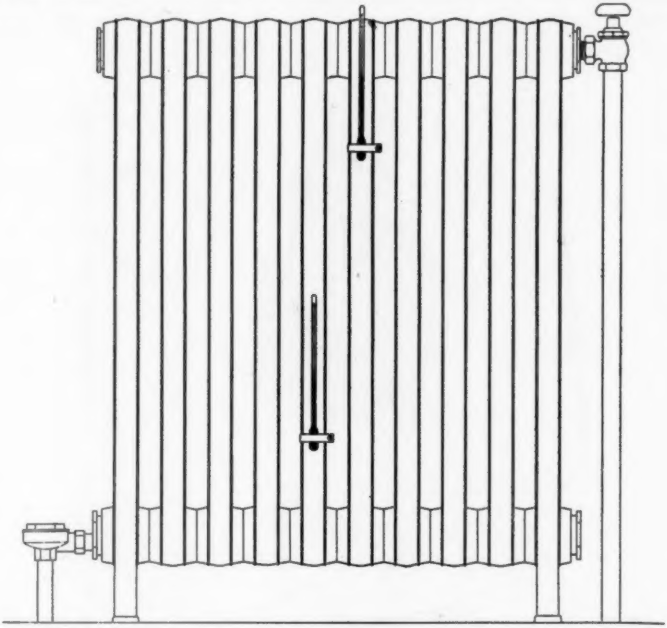

Fig. 2. Typical Arrangement of Thermometers for Methods (2) and (3)

It is very important that the radiation be designed to fit the heat losses and the piping to give even distribution of heat. But even with the heating system balanced for one pressure condition, the distribution may not be equal at all other pressures. Therefore, a temperature regulation system should operate so as to bring about even distribution of heat under all conditions.

This could best be accomplished by keeping the steam lines full at all times and, by means of a controlled valve at each radiator, admitting steam to the radiator as required to keep the radiator at the correct temperature. This method is impractical because of expense. The same results can be obtained by keeping the mains full of steam at the same pressure as in the radiators and, by short bursts of additional pressure, forcing steam into the radiators as

required. By this means the inequalities of piping design are somewhat evened out, and in a fairly well designed system, approximately equal amounts of steam are forced into all radiators under the various conditions of pressure and vacuum.

The adequate temperature control system should eliminate the objectionable conditions of *overshooting* and *lag*. These conditions are bound to occur unless the relation between radiator temperature and outdoor temperature shown in Fig. 1 is maintained at least approximately. *Overshooting* occurs when a radiator, emitting heat at a greater rate than the heat loss, is shut off as the room reaches the correct temperature but, continuing to emit heat at the high rate

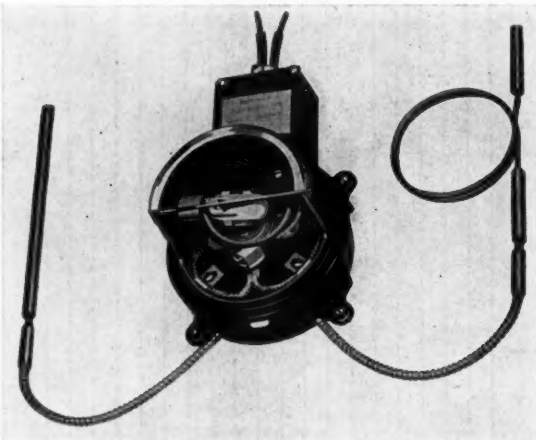

FIG. 3. CUT-AWAY VIEW OF CONTROL INSTRUMENTS

for a period, raises the room temperature excessively. *Lag* occurs when this radiator is turned on again as the room drops to the correct temperature but, since the room cools from the outside wall *in* and since the usual place for measuring room temperature is an inside wall, the room becomes uncomfortably cool near the outside walls before the temperature falls below that required on the inside wall.

It is plain that intermittent heating will produce some *overshooting* and *lag*. If the periods are short and frequent, they will be reduced to a minimum. If the periods are long and infrequent, such as exist in a system controlled by a room thermostat, they are apt to produce uncomfortable temperatures. This will be especially true during the moderate portions of the heating season. However, if frequent periods of intermittent steam supply are used, the relationship indicated in Fig. 1 is approximately maintained, and the *overshooting* and *lag* are negligible.

With these principles of temperature regulation established, the control

apparatus and its operation on a simple steam heating installation will be described to illustrate how the design and operation comply with these principles. Fig. 3 shows a cutaway view of the control instrument. The instrument consists of a case in which is mounted a mercury switch and a Bourdon tube. The Bourdon tube operates the mercury switch and thereby controls the oil burner, coal stoker, gas burner or, in the case of central steam service

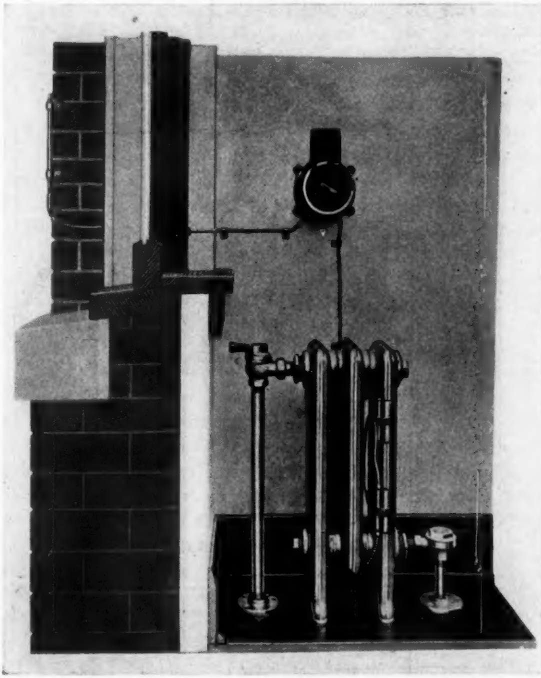

FIG. 4. TYPICAL INSTALLATION OF CONTROL INSTRUMENTS

and zoned heating systems, the steam valve. The Bourdon tube is operated by the expansion and contraction in three metal bulbs which are attached to the Bourdon tube by flexible capillary tubing. One of the metal bulbs is placed outdoors and the other two are attached to a suitable radiator.

It is the total expansion of the liquid in the three bulbs which determines the action of the Bourdon tube and thereby the mercury switch. When the outdoor bulb is warm, the liquid in this bulb will be expanded and just a slight amount of heat applied to the radiator bulbs will be sufficient to move the mercury switch to the off position. When the outdoor bulb is cold, the liquid will be contracted so that it will be necessary that the radiator bulbs be heated to a high temperature in order to move the mercury switch to the *off* position. The instrument will therefore cause the heating unit to operate

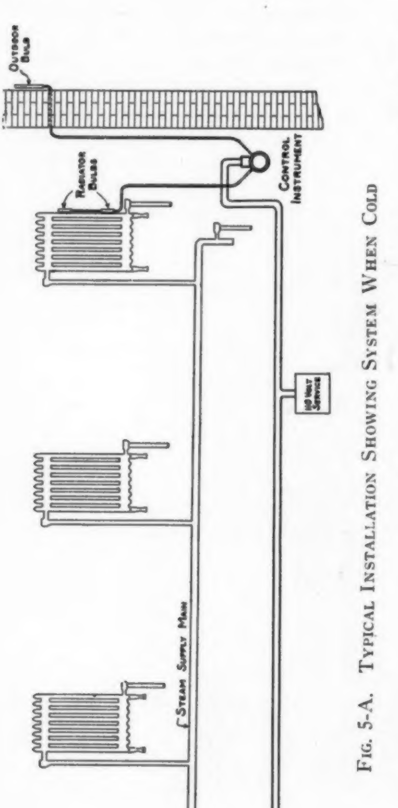

FIG. 5-A. TYPICAL INSTALLATION SHOWING SYSTEM WHEN COLD

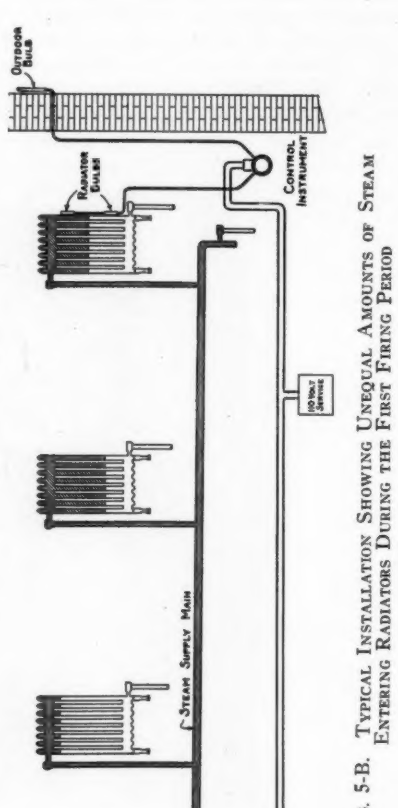

FIG. 5-B. TYPICAL INSTALLATION SHOWING UNEQUAL AMOUNTS OF STEAM ENTERING RADIATORS DURING THE FIRST FIRING PERIOD

in a manner which will produce the desired temperature in the radiator to which its bulb is attached.

For the standard steam heating system the instrument is furnished with an outdoor bulb having twice the volume of the radiator bulbs. Thus each degree drop in outdoor temperature produces a 2-deg increase in the radiator temperature. The instrument is provided with 20 ft of capillary tubing on the radiator bulb and with 10 ft of tubing on the outdoor bulb. A reeess is provided in the back of the instrument to hold any excess of tubing over the length required. In Fig. 3 the tubing is shown coiled up in this recess. The instrument is usually installed in a closet or hallway close to the control radiator with its flexible tubes leading outdoors and to the radiator. Electric wires connect the instrument to the automatic control for the heating plant. Fig. 4 shows a typical installation of the instrument.

In a steam, vacuum, or vapor heating system the radiator bulbs of the instrument are always applied to the last radiator on the last riser on the longest steam main. The reason for using this radiator for the control is to insure a steam supply to all the radiators or at least to the risers leading to all the radiators before the last radiator can be heated sufficiently to shut off the supply of steam.

In order to illustrate the operation of the apparatus, the various cycles of operation in a typical heating plant will be considered. When the instrument starts the steam supply (See Fig. 5A), steam is forced into the piping and the radiators, finally reaching the control radiator on the end of the longest run of pipe. Steam enters the radiator until the radiator has been brought to the correct average temperature, when the instrument stops the steam supply (See Fig. 5B).

During this cycle the steam main has been completely filled with steam, since steam must reach the last radiator before the control operates, but the amount of steam entering the radiators may be unequal, depending on how well the piping is balanced and on the condition of the air valves.

As the control radiator condenses steam and its average temperature drops below the correct level, the control apparatus starts generation of steam. Meanwhile steam has been condensing in the other radiators at a rate proportional to the surface in contact with steam, so that the least filled radiator condenses the least steam. Steam has also been condensing in the mains, but with covered pipes, the simmering of the boiler at off periods will be sufficient to keep them full of steam (Fig. 5C).

As the control apparatus starts the generation of steam for the second time, with the mains full of steam, the increase in pressure is much more even at each radiator, forcing approximately equal amounts of steam into each one (Fig. 5D). Thus with the radiators containing the least steam tending to condense less and the bursts of pressure at frequent intervals tending to send equal amounts of steam into each radiator, there will be a decided tendency for the steam to equalize in the radiators after a few cycles (Fig. 5E).

As before mentioned, the three considerations in the design of an adequate temperature control system are (1) heat emission proportional to heat loss, (2) even distribution of heat and (3) avoidance of *overshooting* and *lag*. The instrument must be rugged and its operation simple.

The apparatus described in this paper provides for the first consideration

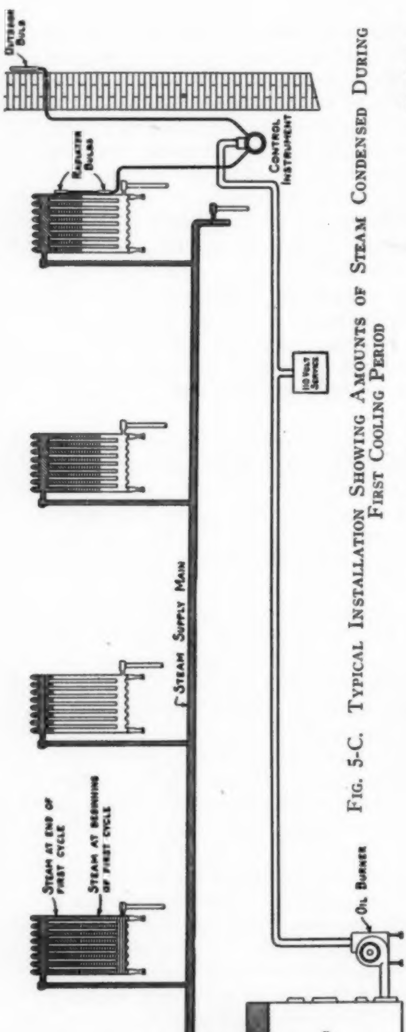

FIG. 5-C. TYPICAL INSTALLATION SHOWING AMOUNTS OF STEAM CONDENSED DURING FIRST COOLING PERIOD

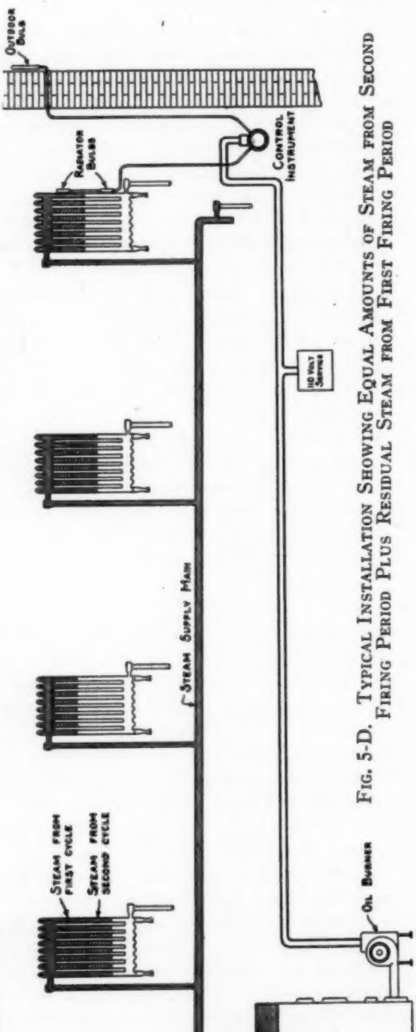

FIG. 5-D. TYPICAL INSTALLATION SHOWING EQUAL AMOUNTS OF STEAM FROM SECOND FIRING PERIOD PLUS RESIDUAL STEAM FROM FIRST FIRING PERIOD

with the counter-balancing action of the outdoor bulb and the radiator bulbs acting on the same Bourdon spring. The problem is solved by going to the real bases, *i. e.,* radiator temperature and outdoor temperature. The second consideration is taken care of by the cycles in the operation of the system, as shown in Figs. 5A to 5E. These cycles tend to bring about more even distribution of steam with each operation, even when the system is not properly balanced. Of course, a balanced system is imperative for the most satisfactory operation. However, the balancing effect of the cycles is always acting at all pressures and vacuums.

The third consideration is taken care of automatically in the solution of the first two. With radiator temperatures held to values equivalent to outdoor temperatures there can be no overheating or underheating. And with the short frequent cycles necessitated by this apparatus the possibility of *overshooting* and *lag* are entirely eliminated.

With this explanation of the design and description of the instrument and its operation, its application to different systems of heating commonly used in large and small buildings can now be described. A modified form of this apparatus is being used for hot water heating systems, which is beyond the scope of this paper. The application to one-pipe steam systems, as used in apartments and commercial buildings, and to vacuum systems, as used in the larger apartments and office buildings, will now be explained. It will be shown how this apparatus operates in conjunction with other control systems, especially with the zone control system, using a specially designed gradual opening valve.

The operation of the one-pipe or two-pipe steam systems controlled by the apparatus is shown in Figs. 5A to 5E. The operation remains the same if there are two or more mains and risers. The control radiator is always the last radiator on the longest run of pipe. If this is on the top floor it is advisable to place the outdoor control bulb on the short mast on the roof. Otherwise it is placed on the outside wall, protected from the sun in all cases.

One-pipe vacuum systems and the various types of two-pipe vacuum systems, with or without pumps, are used to secure better distribution or a more moderate temperature range at the radiator. Thermostatic room control, the maintenance of a constant differential and such devices as modulating valves, return traps, and orifices in the supply line give added refinement in operation. With all of these, this apparatus affords the primary control.

This apparatus is valuable, when used with thermostatic room control, in that it eliminates the *overheating* and *lag*. The thermostat will determine when heat is needed, and the apparatus will determine the rate at which heat is to be delivered. Under this combined control there will be a long and moderated delivery of heat to the room each time the thermostat calls for heat rather than a short and intense delivery. If gradual acting thermostats are used the apparatus is of value as a master control to prevent excessive delivery of heat when tenants throw the thermostats out of operation by opening windows or by abusive manipulation of the thermostats.

In conjunction with the two-pipe vacuum system, the apparatus functions to prevent excessive differential during the moderate part of the heating season. This condition is objectionable because it makes modulating valves, orifices, etc., inoperative. The excess of differential over frictional resistance is caused

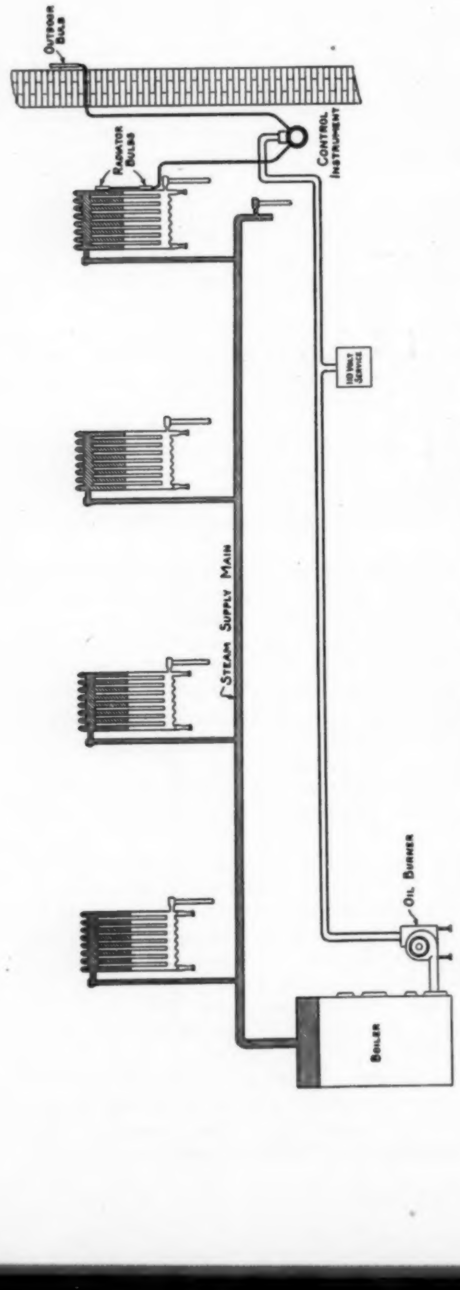

FIG. 5-E. TYPICAL INSTALLATION SHOWING TENDENCY FOR STEAM TO EQUALIZE IN THE RADIATORS AFTER A FEW CYCLES

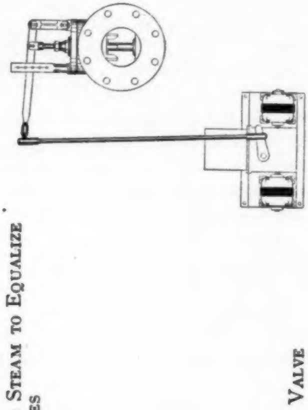

FIG. 6. GRADUAL-ACTING, MOTOR-OPERATED VALVE

by the closing of the return traps on the radiators. With fractional radiator heating afforded by the apparatus, the return traps will remain open until cold weather requires a full radiator when steam pressure can be built up. With the return traps open, the full effect of the return pump is available for

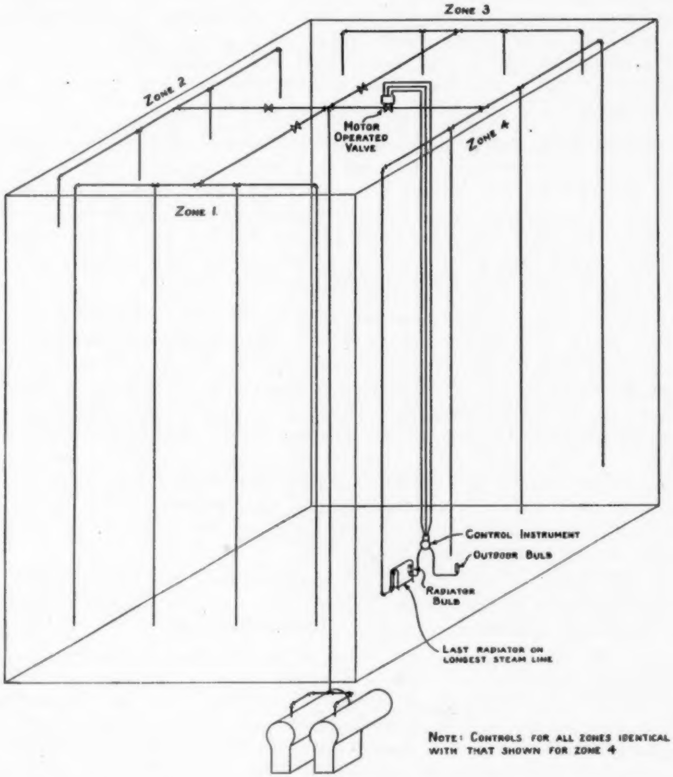

FIG. 7. HEATING SYSTEM DIVIDED INTO FOUR ZONES, ONE FOR EACH FACE OF THE BUILDING

circulation. It is seen that with these systems, the apparatus is the primary temperature control, aided and refined by the other devices.

The zone system of heating is becoming popular because it provides a means of controlling the heat input to various parts of the building to meet the variations in the conditions of exposure. This zone system necessitates the use of valves, either manually or motor controlled, to adjust the flow of steam to each zone, and some method of control for these valves. A gradual-acting, motor-operated valve has been developed to operate for zone control in con-

junction with the apparatus described in this paper. The valve is shown in Fig. 6. This valve is operated by two motors, one to close and one to open, and the time required from full open to full closed, and from full closed to full open is 20 min.

Fig. 7 shows a heating system divided into four zones, one for each face of the building. Steam is supplied to each zone through a gradual-acting, motor-operated valve. Each valve is controlled by the apparatus connected to the control radiator in each zone. This is the radiator farthest from the supply valve. The valves can also be operated from a switchboard in the boiler room for night operation and morning pick-up.

Each zone, from supply valve to control radiator, acts in practically the same way as the simple system shown in Figs. 5A to 5E. Due to the long closing and opening time of the motor-operated valves, the changes in temperature of the control radiator will occur before the valve is completely closed or opened. This will cause the valve to operate from one fractional position to another rather than from full open to full closed. The steam supply will therefore be continuous under a slightly oscillating pressure.

The outdoor bulb for each zone is placed on the face of the building which that zone heats. The bulb is protected from direct sunshine by a shield but is open to circulation of the *skin* air of the building. High wind will wash the face of the building thereby producing a lower *skin* or surface temperature. On that face of the building where there is no wind there will be a rising current of air along the face of the building due to the heating effect of the building. This produces a higher *skin* temperature. Similarly, sunshine on one face of the building will produce a higher *skin* temperature. Thus the heat supplied to each zone is controlled by the *skin* temperature of the building in that zone, which automatically compensates for the variables of wind and sunshine, as well as for outdoor temperature.

With this system of zone control, the rooms on each face of the building are supplied with heat in the amount indicated by the *skin* temperature on that face. Each zone operates as a complete, independent unit.

It has been shown by the description of its operation in conjunction with the various types of heating systems how this control apparatus will maintain a comfortable temperature in all parts of a building. By the elimination of *overshooting* and *lag,* this comfortable temperature will be lower than in the presence of these two factors since a room which has been at a temperature higher than 70 F is no longer comfortable as it cools to 70 F. Therefore, a fuel saving is obtained as well as a more equable temperature.

DISCUSSION

T. F. McCoy: Does the author consider it essential to use orifices with this system of regulation, in order to produce an even distribution of steam? When the main valve opens and lets a jet of steam into those radiators nearest to it the first rush of steam will overheat those rooms, whereas, the radiators in rooms at the far end of the line will lag behind and continue to call for heat.

R. A. Wolff: I understand that there is a method of operating this system

with a thermostat, so that a combination of exterior and interior temperatures can be used, but it has never been clear to me.

S. R. LEWIS: My major criticisms of this paper are favorable, because I know that the author has solved many troublesome problems of automatic temperature control.

I. S. DANE: I would like to inquire about the uniformity of control when some rooms have extended surface radiation and others have cast-iron.

H. M. HART: This discussion indicates the possibility of eliminating some of the difficulties experienced with oil and gas burners. Today our heating systems are generally designed solid fuels and for constant heat supply through the system. When intermittent firing is used, heating contractors have a lot of trouble, but this type of control should encourage the greater use of oil and gas-fired installations.

MR. LEWIS: With a modern oil-fired vapor steam heating system having graduated control valves on each radiator, and a single thermostat located in some representative room, frequent complaints result. Because the vapor system is especially designed to give optimum localized control and the central thermostat attempts to govern the temperature on an entirely different principle, a troublesome condition is created.

Apparently central thermostatic control for a house is better adapted to warm-air, hot-water or air vented steam systems than to vapor or vacuum systems equipped with graduated manually controlled supply valves on the radiators. This new control tends to overcome objections to such a hook-up.

In most cases, however, when we install an elaborate local temperature control at the radiators and add a centrally located thermostat which attempts to control the whole house, we must expect to receive complaints that one or the other of the two schemes for control is not good.

For vapor systems I am inclined to recommend that the oil burner shall be controlled from the steam temperature rather than from the room temperature so that the plant can be ready instantly to serve any radiator valve which may be opened.

W. T. JONES: We have used this device on oil burner installations and we find that it has cut down materially on the amount of oil that has been burned.

MR. GATZENBERGER: I am wondering whether there is a limitation on the length of capillary tubing, also whether this is a liquid filled or gas filled system? The one shown here indicates that the size of the bulb, in relation to the volume of capillary tube, is small. If it is a question of liquid filled or gas filled system, I would say that on the effect of temperature along a capillary tube would upset greatly the outdoor effect on the bulb.

MR. APMANN: In the last year this type of heat regulation has been used in several types of buildings. The first installation was in an apartment building in which recording thermometers showed that at 78 deg the occupants reported that they were uncomfortable. The installation with the new control system showed a 73 deg temperature on recording charts which was reported as entirely comfortable, largely due to the psychological effect of having warm radiators at all times.

One of the tenants told me the other day that he sits and watches the curtains move and all of a sudden he can see them move a little faster and then slow

down, indicating the success with which the control regulates the flow of heat to the radiator and, of course, increases the comfort.

In a 9-story office building where both concealed radiation and ordinary standard radiation were installed the temperature was uniform throughout the building and the fuel consumption decreased.

F. I. RAYMOND: Regarding the necessity of using orifices we find them of great benefit in balancing the system, in meeting the early morning warming-up demands. We find that we can control a one-pipe steam heating system quite satisfactorily due to an inherent balancing effect of the system. The paper contains a more theoretical explanation of the operating characteristics. If steam is present at all radiators, but not flowing into them, whenever steam pressure is raised the quantity of steam flowing into each radiator is proportional to the length of time that the pressure is maintained. That is because the air escapes through the valves slowly and not all at once. Consequently, if steam pressure is raised for five minutes, it tends to fill one-quarter of each radiator with steam. Of course, if one valve is plugged and another valve is wide open, the heat will be uneven.

Mr. Wolff asked about the hook-up with the thermostat. In the case of an electric system the connections to the thermostat are made just like those to an aquastat or to a pressure control, so that either the control or the thermostat can shut off the fire. Because the radiator temperature always rises before the room temperature, the control bulb will be affected before the room temperature rises so that the fire is shut off anywhere from 10 to 15 min earlier than would occur with a thermostat. The statement should be qualified to this extent, in very small homes with a thermostat properly located the temperature regulation is excellent. I am frank to say that in a very nicely laid out six-room home I could detect no appreciable difference between the combined operation of this control and the thermostat and a thermostat alone.

Mr. Dane asks how the control works with a combination of extended-surface and cast-iron radiation. There are no difficulties involved because steam is always present in the main with a gentle flow into the cast-iron radiation as well as into the extended-surface radiation, in the first a living room and the second bedroom, respectively.

A comment I should like to make on Mr. Lewis's remarks is that I was entirely sold on hot-water heat until I found out how much the circulation would vary in the piping system. The balance is almost as difficult to obtain as in a vapor system. I agree with Mr. Lewis's opinion that a vapor system should not be controlled by a thermostat.

Mr. Gatzenberger will find a description of the instrument in this paper. The instrument, completely liquid-filled, is regularly provided with 30 ft flexible tubing and the outdoor bulb is ⅜ in. in diameter and 8 in. long. It is assumed that the capillary tubing will have the same temperature as the instrument, but there is compensation in the instrument to take care of a temperature difference.

No. 863

FRICTION LOSSES AND OBSERVED STATIC PRESSURES IN A DOMESTIC FAN FURNACE HEATING SYSTEM

By A. C. Willard[1] and A. P. Kratz,[2] Urbana, Ill.

MEMBERS

Acknowledgments

THE data presented in this paper were obtained in connection with an investigation which is being conducted by the Engineering Experiment Station of the University of Illinois, of which M. S. Ketchum, Dean of the College of Engineering, is Director, in co-operation with the *National Warm Air Heating Association.* This paper constitutes part of a more complete report on the further investigation of warm-air furnaces and heating systems which will appear as a future bulletin of the Engineering Experiment Station in the furnace heating series. Much of the actual experimental work involved in this study of fans as carried out in the Research Residence plant was performed by J. F. Quereau, formerly Special Research Assistant attached to this investigation.

Under the terms of a cooperative agreement between the *National Warm Air Heating Association* and the University of Illinois, a very extensive study of furnace heating problems has been made, using first an experimental plant with auxiliary equipment in the laboratory, and later a typical modern residence erected by the association for the express purpose of correlating and extending the work in the laboratory to the conditions of actual installation. It was in this residence that the data herein presented were obtained.

Introduction

The increasing tendency to apply various types of small disc and propeller fans as well as small centrifugal fans to warm-air furnace heating systems of the gravity circulating type has developed without very much attention to the heads or resistances which these fans must overcome. Little consideration has been given to the unusual operating conditions surrounding such a fan installation, since these fans must operate in a more or less rapidly moving current of air created by the gravity or natural circulation of the furnace. The rating of a

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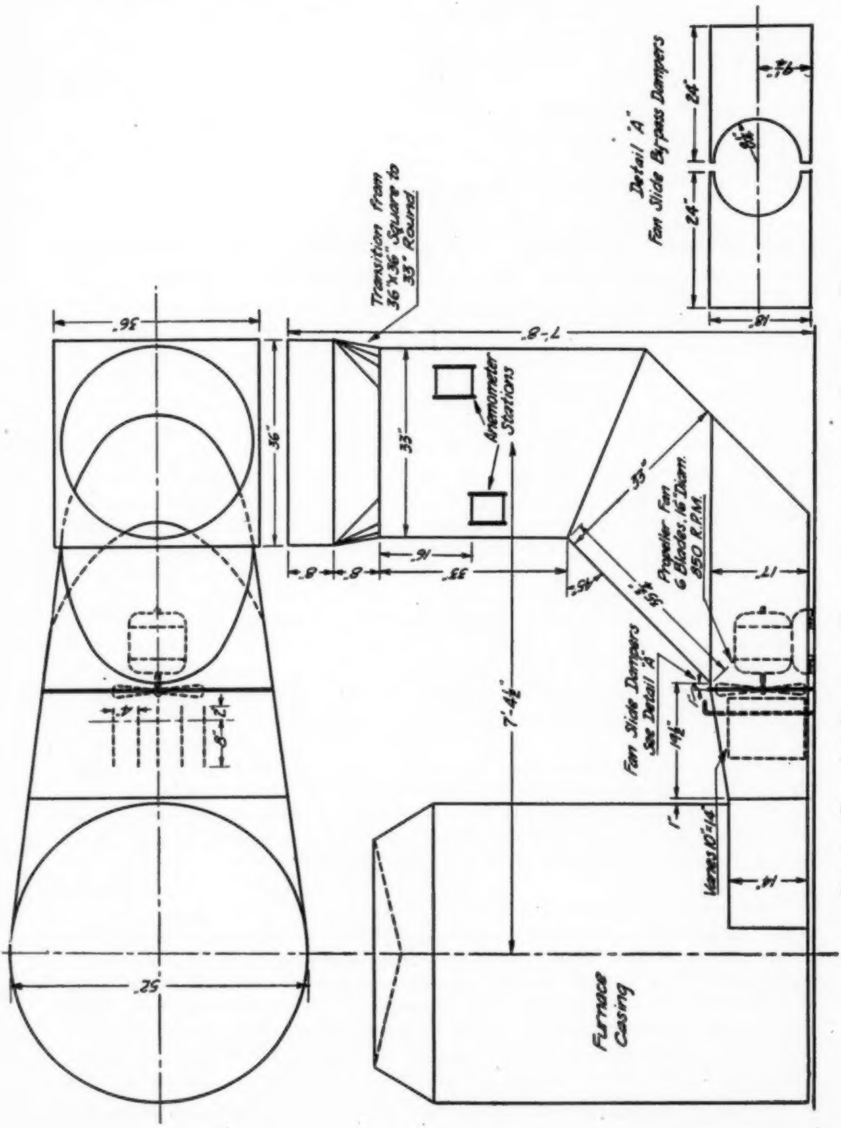


FIG. 1. COLD AIR DUCT FOR NINTH INSTALLATION IN RESEARCH RESIDENCE

fan under the usual *still air* conditions *does not apply* when the fan is installed within a duct and in a *moving air* stream of variable velocity and of a volume which is often equal to or even in excess of the fan capacity in *still air*.

As a result of the failure to take account of the peculiar conditions surrounding the installations of fans in domestic furnace heating systems, the following possibilities or situations may arise:

(1) With a well designed gravity circulating system and the usual commercial fan as applied today, the natural circulation of the furnace may practically submerge the fan and the fan may become a drag on the system. This is almost certain to occur at high combustion rates, and with small fans.

(2) With any system designed for gravity circulation the fan will almost invariably unbalance the previous gravity operation of the system, whether good or bad, and tend to increase the flow in those pipes in which there was the least resistance and friction as a gravity system. This means, usually, more air to first floor rooms or to short direct runs; and relatively less air to distant rooms. Such a condition may defeat the objective of the fan installation entirely, as shown in item (3).

(3) Since, as explained in item (2), the more air is certain to go to the favored than to the unfavored rooms, the former will immediately overheat unless the air temperature at the registers to these rooms is reduced. The occupant or the thermostat on the first floor will promptly check the fire and maintain 70 F in the favored rooms with the result that the unfavored rooms may actually drop below 70 F and cool off.

(4) Since both the fan and the natural circulation always take place simultaneously when the furnace is under heat, the resistance which the fan has to overcome is an extremely variable quantity, which *may actually range from a positive to a negative value*. In all cases, it will be extremely small; ranging (with the usual type of propeller fan) in a well designed system from a static pressure in the bottom of the furnace casing of +0.010 in. of water for cold furnace with air at 65 F to —0.008 in. of water for hot furnace with air at 150 F at the registers. The fan tested had a *free air* capacity of 1,400 cfm and actually delivered 950 cfm as installed in the furnace shoe with by-pass dampers closed. Even with a centrifugal fan delivering 1,260 cfm of air, the maximum static pressure in the bottom of the casing was only +0.02 in. of water with cold furnace at 65 F and the minimum 0± inches with hot furnace at 150 F.

(5) The propeller type of fan placed in the return duct develops a greater air handling capacity with the by-pass dampers closed than with them open. In either case, the fan suffers a material reduction in its *free air* handling capacity. The net aspirating effect of such a fan in a large return duct with dampers open is of negative value since any induced flow which may occur is much more than counterbalanced by the eddy currents and short circuiting which can and does occur when there are no by-pass dampers. Such fans should, therefore, be installed as is generally customary with automatic by-pass dampers when placed in large return air ducts.

(6) Noise of a more or less noticeable character is almost certain to be transmitted to the occupied rooms of the residence. The occupants are never in doubt as to whether or not the fan is in operation.

In addition to the preceding discussion of pressure effects on the performance

FIG. 1. COLD AIR DUCT FOR NINTH INSTALLATION IN RESEARCH RESIDENCE

of small fans when installed in the return ducts of warm air furnace heating systems, it should be made clear that a fan of proper size may have some beneficial effects on residence heating. For example, provided the furnace is hot and the fire is maintained in normal condition, the starting of the fan will almost instantly increase the heating capacity of the furnace and its efficiency. The increases in the system under consideration were large and, although they decreased rather rapidly within half an hour, some increase persisted even at the end of two hours *provided the same intensity of firing was maintained.*

Typical increases in heating capacity were as follows:

(1) With furnace delivering 18,000 Btu per hour at bonnet and a low fire,

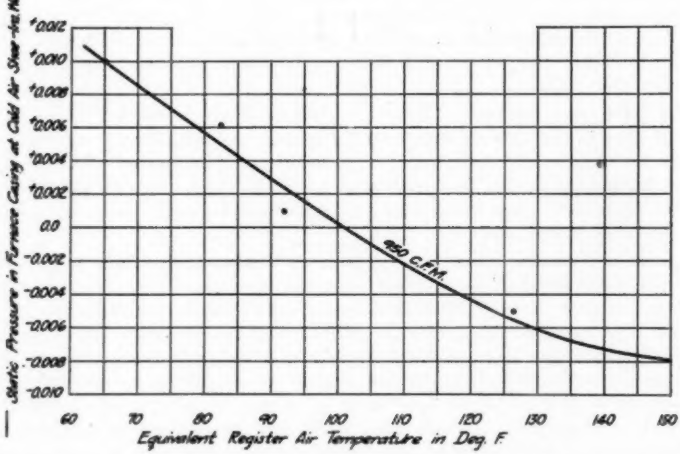


FIG. 2. STATIC PRESSURE IN FURNACE CASING AT COLD AIR SHOE. RESEARCH RESIDENCE—9TH INSTALLATION. 6 BLADE PROPELLER FAN, BY-PASS DAMPERS IN. VOLUME OF AIR BASED ON TEMPERATURE AT 65 F

starting the fan increased the heating capacity to 33,000 Btu in a minute and a half, with a drop to 27,000 in 30 minutes, which became 24,000 in two hours.

(2) With furnace delivering 62,000 Btu per hour at bonnet and a brisk fire, starting the fan increased the heating capacity to a maximum of 83,000 Btu in a minute and a half, with a drop to 73,000 in 30 min which became 70,000 in two hours.

Since the combustion rate was maintained constant in each case the furnace efficiency was benefited in almost the same degree as the heating capacity of the furnace.

The fact must not be lost sight of that starting a fan will do little good unless there is a residual heat in the furnace itself and an active fire is maintained. Fans should not be depended upon to correct improperly designed or poorly

installed gravity circulating furnace heating systems. If the system is already out of balance, the fan may merely aggravate the situation unless the distributing system is readjusted to favor the unfortunate rooms. A properly designed and correctly installed gravity circulating system does need a fan.

DESCRIPTION OF APPARATUS

The equipment used for these tests was the heating plant installed in the Warm Air Research Residence, which has been completely described in University of Illinois Engineering Experiment Station Bulletin No. 189.[3] This plant consisted of a 27-in. cast-iron circular-radiator furnace equipped with a

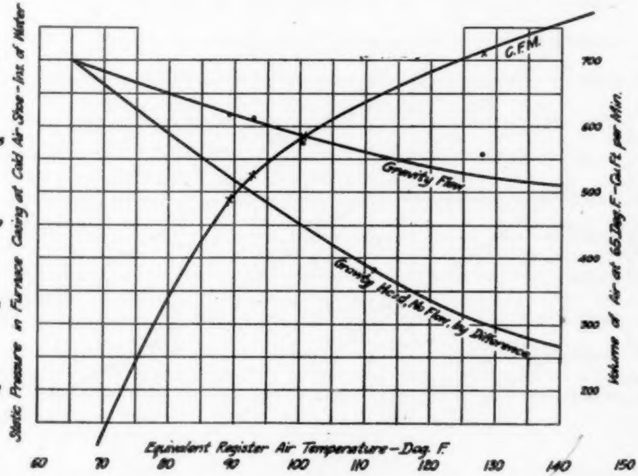


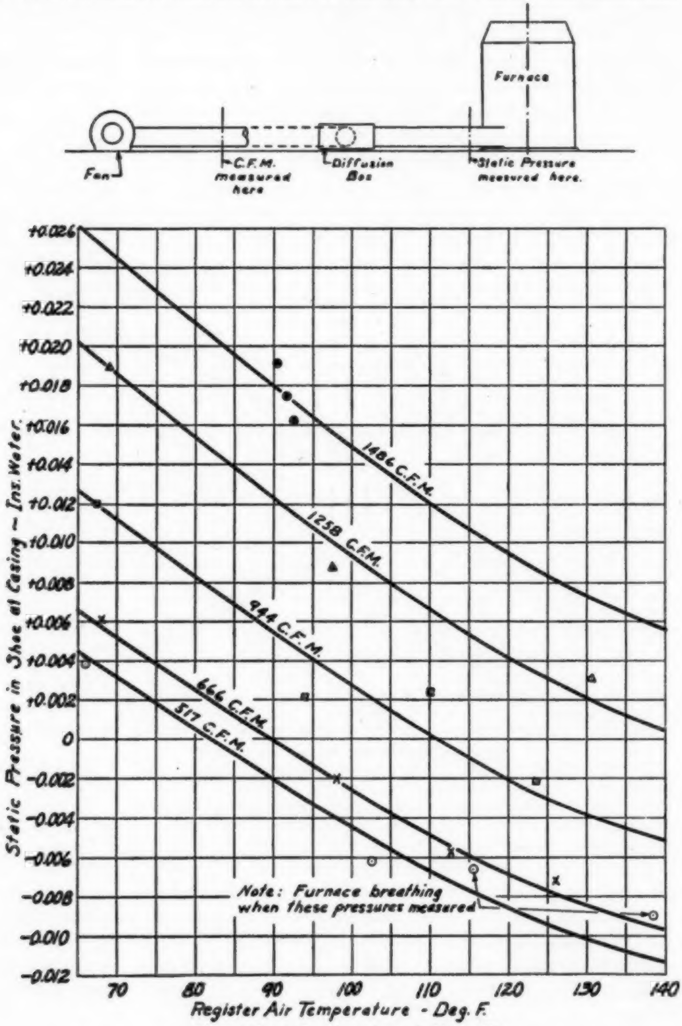
FIG. 3. STATIC PRESSURE AND AIR VOLUME AT COLD AIR SHOE. RESEARCH RESIDENCE, 9TH INSTALLATION. GRAVITY FLOW. TEMPERATURE OF AIR 65 F

52-in. casing, supplying air to 11 basement warm air pipes having a cross-sectional area of 832 sq in. When equipped with a single recirculating duct of 854 sq in. cross-sectional area, this furnace heated the house in zero weather with register air temperatures approximating 135 F. Under these conditions practically 680 cu ft of air based on 65 F and 29.5 in. of mercury were circulated per minute.

No changes were made in the casing or in the warm air side of the system. Each change in the cold air side of the system has been designated by an installation number, and this discussion is confined to the seventh and ninth installations.

For the seventh installation, the cold air return duct was replaced by a shoe and a diffusion box shown in Fig. 4. The shoe was 12 in. in height and the

[3] "Investigation of Warm-Air Furnaces and Heating Systems, Part IV, Research Residence," by A. C. Willard, A. P. Kratz, and V. S. Day, Engineering Experiment Station Bulletin No. 189.



Shoe 12" x 39" Shoe Area 3.25 Sq Ft
Volume and Velocity of Air Based on Temperature of 65 F
Average Temperature of Air Inlet 65 F

● =	Gate Position # 1;	Cu Ft per Min	— 1486;	Velocity Ft per Min.	— 457
△ =	" " # 6;	" " " "	— 1258;	" " " "	— 386
□ =	" " # 8;	" " " "	— 944;	" " " "	— 290
× =	" " # 9;	" " " "	— 666;	" " " "	— 205
○ =	" " #10;	" " " "	— 517;	" " " "	— 159

FIG. 4. STATIC PRESSURE IN SHOE WITH FAN FURNACE. RESEARCH RESIDENCE, 7TH INSTALLATION, MULTIBLADE CENTRIFUGAL FAN

sides were tangent to the furnace casing. A multiblade centrifugal fan delivered air through a 12-in. round pipe into the side of the diffusion box in such a manner that the velocity of the jet was destroyed, and the air was delivered uniformly across the cross-section of the shoe.

In the ninth installation, a single cold air return duct with a cross-sectional area of 854 sq in., as in Fig. 1, was used. This duct was connected to a shoe similar to the one used in the seventh installation. A 6-blade propeller fan 16 in. in diameter was installed in the shoe, and slides, or dampers, were provided so that the fan blades revolved in an orifice with a clearance of ¼ in. These dampers are shown in the detail in Fig. 1. When the furnace was operated under gravity flow, the dampers were removed so that the full cross-sectional area of the shoe was effective.

Test Procedure

For the seventh installation the volume of air delivered was measured by means of a Pitot tube placed in the 12-in. round pipe 8.5 ft from the fan discharge. A 10-point traverse on each of two diameters was used. The angle between the diameters was 90 deg. The static pressures were observed by making a traverse of the shoe with a Pitot tube placed successively in six positions in the cross-section just outside of the line of the furnace casing. The Wahlen gage, or Illinois micromanometer, reading to 0.0005 in. of alcohol was used to measure the pressure. Pressures above atmospheric have been designated as plus and those below atmospheric as minus.

For the ninth installation, the volume of air was measured by means of a calibrated anemometer in the recirculating duct. For this purpose, a traverse was made on two diameters at 90 deg to each other. Six points were used on each diameter. Since considerable jet action was found in the shoe itself, indicating that the shoe did not *run full,* the static pressures in this case were measured by means of a Pitot tube inserted in the casing about 3 in. above the top of the shoe. In this plane, practically all of the velocity of the jet from the fan had been converted into pressure head. These pressures were also read by means of the Wahlen gage.

In all cases, the plant was maintained at a predetermined register air temperature for a period of sufficient length to establish thermal equilibrium, and this temperature was held constant while the pressure measurements were being made.

Discussion of Results

The propeller fan delivered 950 cu ft of air per minute, and this amount remained practically constant over the whole range of register air temperatures. The observed static pressures with the propeller fan in the 9th installation are shown in Fig. 2. The static pressure corresponding to a register air temperature of 65 F represents the head necessary to overcome the friction in the warm air side of the system including the registers, wall stacks, basement pipes, casings, and shoe between the fan and the casing. It may be noted that this pressure did not exceed 0.01 in. of water, with a cold furnace.

As the register air temperature was increased, the increasing negative, or suction, head created by the furnace combined with the positive pressure head created by the fan and resulted in decreasing values for the observed static

pressure. The latter became zero at a register air temperature of 100 F, and was negative for all register air temperatures greater than 100 F.

Since the observed static pressures at all register air temperatures above 65 F represent the composite effect of the pressure created by the fan and the suction created by the furnace, it is reasonable to assume that the suction head produced by the furnace is represented by the difference between the observed static pressure for the given register air temperature and the observed static pressure at 65 F. These differences are shown in the lower curve in Fig. 3. The curve for volume of air circulated per minute under gravity flow alone is also shown in this figure.

The upper curve in Fig. 3 represents the observed static pressures, or suc-

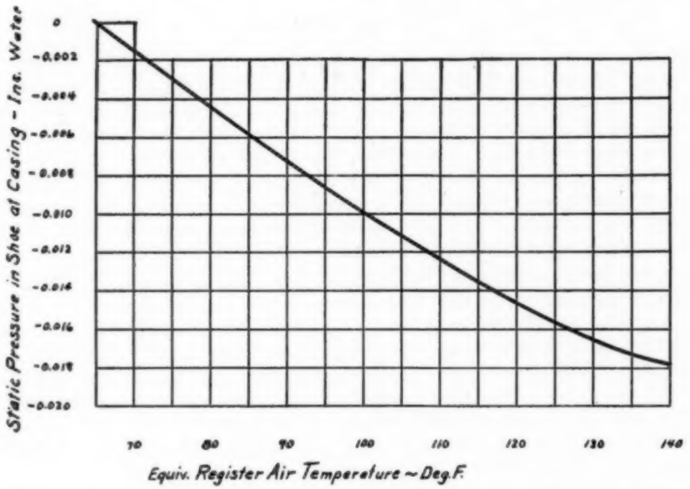


FIG. 5. STATIC PRESSURE IN COLD AIR SHOE. RESEARCH RESIDENCE—7TH INSTALLATION. MULTIBLADE CENTRIFUGAL FAN. GRAVITY HEAD, NO FLOW, BY DIFFERENCE. AVERAGE TEMPERATURE OF AIR AT INLET 65 DEG. F

tion heads produced by the furnace under the conditions of gravity flow with the fan not in operation. It may be observed that these suction heads are less than the ones determined by difference, and shown in the lower curve. When the furnace is operating under its own motive head, it must create not only the head necessary to overcome friction but also the velocity head. Hence the observed static pressure will be less than the theoretical head calculated from the difference in weights of the air columns on the hot and the cold sides of the system.

When the fan is operated, it creates the velocity head and probably either all, or part, of the head required to overcome friction. The gravity head obtained by difference, shown in the lower curve, therefore, probably represents the theoretical gravity head, or the gravity head under conditions of no

flow. An approximation of the latter may be made by application of the chimney formula:

$$D = 7.64H \left(\frac{1}{T_a} - \frac{1}{T_c}\right)$$

in which D = theoretical draft in inches of water, H = height of chimney in ft, T_a = absolute temperature on cold side of the system in degrees Fahrenheit, and T_c = absolute temperature on hot side of the system in degrees Fahrenheit. In applying this formula, it was assumed that the height of the second story wall stacks from the grate line to the centers of the registers represented the mean height of the warm air columns, and the average register air temperature represented the mean temperature on the hot side. This com-

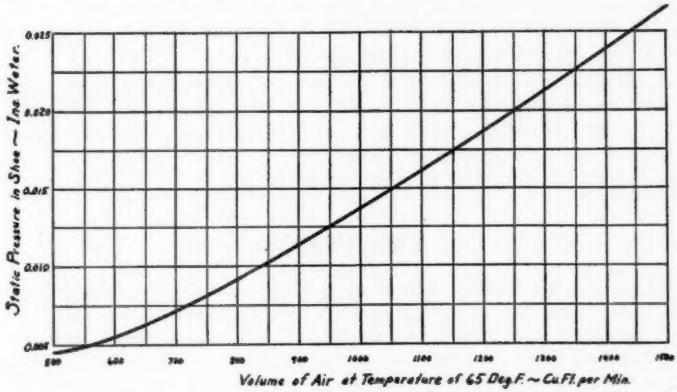


FIG. 6. STATIC PRESSURE IN COLD AIR SHOE. FAN RUNNING IN COLD FURNACE. TEMPERATURE OF AIR 65 F. RESEARCH RESIDENCE, 7TH INSTALLATION MULTIBLADE CENTRIFUGAL FAN

putation resulted in, $D = -0.0267$ for a register air temperature of 130 F and $D = -0.0151$ for a register air temperature of 100 F. The corresponding values read from the lower curve in Fig. 3 were —0.0165 and —0.0100 respectively. This agreement is reasonably close, considering the uncertainty in regard to the actual mean height and actual mean temperature for the system, and tends to confirm the conclusion that the suction heads determined by difference represent the gravity heads for no flow.

Fig. 4 shows the observed static pressures determined for various register air temperatures and volumes of air circulated with the multiblade centrifugal fan in the 7th installation. These curves also indicate that the observed static pressures decrease as the register air temperatures increase, and in addition that the observed static pressures increase as the volume of air circulated increases. Some inconsistency may be observed in two of the lower points obtained with very low air volumes and high register air temperatures, because under these conditions the furnace *breathed* and air was drawn in through the warm air register in the kitchen.

Fig. 5 shows the theoretical or gravity head under no flow conditions, and exhibits the same characteristics as the lower curve in Fig. 3. The numerical values also agree very closely at given register air temperatures for the two curves.

Fig. 6 shows the static pressures or friction heads for the cold furnace for various volumes of air circulated. The maximum value of 0.025 in. of water was obtained when the volume circulated was practically twice the normal gravity air circulating capacity of the furnace in zero weather. Furthermore, the value of 0.012 in. of water for 950 cfm agrees very closely with that of 0.010 shown for an equivalent register temperature of 65 F in Fig. 2, in which case 950 cfm were circulated by the propeller fan. A somewhat higher pressure is to be expected in the case of the centrifugal fan, since the pressures were measured just outside of the casing and hence included the entry loss for the shoe. The pressures measured in the casing, just above the shoe, with the propeller fan did not include this loss.

The outstanding feature of all of the curves is the fact that the friction loss in a well designed gravity furnace system is small and that the observed heads are very low even with a cold furnace. Furthermore, when the furnace is under heat, these heads become progressively smaller, and assume negative values at the higher register air temperatures.

DISCUSSION

V. S. DAY (WRITTEN): Referring to Conclusion 2—the inference may be drawn that where fans are employed the distribution must be maintained by proper adjustment of dampers, or by the introduction of proper resistances in the individual pipes. In either case the distribution will not be satisfactory for gravity operation. Recent experience has indicated that predetermined orifices may be satisfactorily used as a means of converting the pipes of a gravity warm air system to the purposes of forced flow.

The term, "relatively less air," in Conclusion 2 is not clear. Is it not true that when the fan was applied, the air to each room was increased in about the same proportion, and could not the over-heating of certain rooms be the result of increased air quantity rather than in any disproportioning of the percentage sent to the various rooms?

Conclusion 6 should hardly be construed to mean that effective sound deadening cannot be obtained. Recent results show that in systems designed for forced air flow, quietness to a degree inaudible to the human ear can be obtained, when proper acoustical methods are employed.

It would be most desirable for the authors to give the Society the results of their observations on the difference in temperature distribution in the rooms heated with and without the forced flow of air.

O. K. DYER (WRITTEN): It is not my intention to question or criticize the method employed in testing, or the results obtained, but I wish to emphasize the advantages to be gained by using a fan with a warm air furnace, and point out that in my opinion the results of the tests do not justify the conclusions reached in the paper.

A more rapid circulation of air around the furnace gives a more nearly

uniform temperature in all rooms, because a large volume of warm air is supplied instead of a small volume of hot air. Also stratification is prevented and a higher temperature is maintained at the breathing zone.

Tests have shown that in the average home without a fan the temperature varies approximately 1 F for every foot elevation from the floor to the ceiling, while with a fan properly installed and operated this temperature difference is reduced ½ F within a few minutes after the fan has been started.

Some rooms in a large number of furnace heated homes do not get their full quota and I have seen many where the hot air riser acted as a return air duct, due to small risers or long horizontal runs, and about the only corrective is to install a fan.

After adjustment of the volume of the dampers which control the amount of air delivered to each room is made no further attention is required and I have never seen a case where a fan used under such conditions failed to improve the heating. After the fan has been operated for a sufficient time to heat up these obstinate pipes it may be stopped and good circulation will continue for some time.

The fan used in the test had too many blades, in my opinion. Two or three would give a greater capacity under the slight pressure required and would offer less resistance to gravity flow when the fan was stopped, and in addition would be quieter in operation.

The dampers used in the tests on each side of the fan were manually operated, which would be impractical for the average home. As automatic dampers depend upon pressure in the furnace chamber to close them the fan starts up, more power is required and this produces more noise. Dampers properly made for use with a fan are not required with a correctly designed cold air duct.

The cold air duct on the test installation was designed for gravity circulation, and is doubtless properly proportioned for that purpose but it is not right for a fan installation. The duct should have been about 40 per cent higher and 50 per cent narrower, then there would have been no eddying or short circuiting.

The figures given in this paper show that the capacity of the plant and also the efficiency was nearly doubled when the fan was operating. It seems remarkable that the fan could practically double the volume of air passing through the furnace, of this model gravity installation which was designed to give full capacity without the air of a fan. Very few such installations are to be found in homes. Generally a local tinsmith uses his own ideas about proportioning the air ducts, so that the fan ought to show to even greater advantage with the average job.

A variable speed, choke-coil, or condenser type motor is recommended, as the resistance, capacity, etc., is rarely the same on two installations. Such motors do not interfere with radio reception, permit speed adjustment to fit the job, and reduce noise to the minimum. The average cost of operating the motor is less than 50 cents per ton of coal burned.

In summary of the paper the statement is made that, "fans should not be depended upon to correct an improperly designed, or poorly installed gravity system . . . that the fan may aggravate the situation, unless the distribution is readjusted." The average householder would conclude from the foregoing that using a fan would not solve his problem. However, this readjust-

ment is not hard, no pipes need to be changed—just partially close the volume dampers in the air supply pipes leading to the most favored rooms and it will be found that the efficiency of the furnace has been increased about 50 per cent.

In the same paragraph the statement is made that, "a properly designed and correctly installed gravity system does not need a fan." Yet these tests made on a model system which must have been the last word as to proper design and installation showed a permanent increase in both capacity and efficiency of about 30 per cent and a temporary increase of over 50 per cent.

The natural conclusion is that the results fully justify a fan on any furnace job.

J. C. MILES (WRITTEN): This paper, which includes a general discussion on performance of gravity and fan systems is, in my estimation, a very important paper, and merits profound consideration.

It is significant at this time, in that it pertains to the air heating principle and as air heating is one phase of air conditioning, the subject matter becomes very important, especially because of the fact that air conditioning is considered vital to the health and comfort of all housed people, particularly those in the home. The apparatus used in this test is simple in its construction, nominal in cost and, therefore, is of interest to the public.

I should like to discuss some portions of this paper for the purpose of clarification.

In items 1 to 6 of the introduction reference is made to promiscuous use of small fans (commercially called—booster fans) without regard to conditions surrounding their installation. If the qualification applies to all items, I readily agree, because my experience has shown that without a material increase in air volume, and a subsequent increase in impingement and static pressure in the heating chamber, little or no improvement may be expected. I would, however, question the broad statement that, "a properly designed and properly installed gravity system does not need a fan." If this statement is meant to apply to all fans on gravity warm air furnaces, I consider it superfluous, in that, "a properly designed and correctly installed gravity system," is a remote possibility, in view of the many adverse conditions encountered when installing the average system. If this is a conclusion drawn from the data given in the paper, I think it is in error.

In the complete report of the furnace investigation referred to in this paper, Fig. 12 shows efficiency and capacity curves for both gravity and fan performance, wherein the efficiency for gravity and fan performance is shown as 58 per cent, and for fan performance 73 per cent. At 3.6 lb combustion rate, the capacity is shown as 81,000 Btu for gravity performance, and 98,000 Btu for fan performance, with the indications that the increase becomes greater as the combustion rate increases, which in turn would indicate a greater increase in the usual application of the system. There is, however, an equalization as well as a reduction shown for the ninth and tenth installations both of which obviously had wrong fan applications.

These curves are for bonnet or register performance and evidently the conclusion was not drawn from these facts.

The over-all or house efficiency may justify the conclusion if we were to view the subject from a restricted perspective. If all chimneys were in the

center of the building as the one in this case, naturally a portion of the heat we are not able to get into the air with the gravity system would be absorbed by the inside walls, but the questions arise: What portion of the building do we want heated? By heating what proportion of the building will we improve our conditions? Judging from present conditions, is there any need of increasing the inside wall temperature?

Personal observation and investigation recently made by Professor Willard at the University of Illinois leads me to believe that it is much more important to improve the outside wall surface temperature. In view of the fact that this paper does not account for an increase of 20 per cent of the heat delivered at the register, I am inclined to think that the increased air currents from the fan, greater impingement on the outside walls and this unaccounted for heat, in part, at least, becomes useful in outside wall surface temperature, therefore, should be regarded as an improved condition.

Another observation I want to make is that the comparison made between the fan system and the gravity system is not a fair one, because this test was made with the very best possible gravity system and the worst possible fan system.

I have had a great deal of experience with alternating gravity and mechanical furnace systems, utilizing the automatic by-pass dampers, having introduced the idea to the furnace industry in 1918 and since that time I have been in continuous contact with installations, good, bad, and indifferent, and I believe that the unbalanced conditions, referred to in this paper, are due to the fact that this experiment was made with a system designed to cope with special characteristics peculiar to strictly gravity circulation. I think that if the assistance of intermittent fan circulation was adhered to, in the original design, the results would be much better and would be quite satisfactory for general use.

Fig. 22 shows a material increase in the fan system capacity for a short period, over gravity capacity, due to the accumulated heat in the heater castings. If my knowledge of physics serves me well, I would conclude that during the period of conductivity in the casting, the heater is working at its very highest efficiency and from this, conclude also that if these periods can be made frequent, the general efficiency would be increased in direct ratio.

In view of the facts set forth in the complete report, I cannot reconcile the statement that a fan is not needed. In fact, this statement would appear to contradict the facts set forth in the report and depreciate obvious advantageous results shown by the tests. Surely it is an advantage to increase the heating capacity of a warm-air furnace from 18,000 Btu to 24,000 Btu with no increase in fuel consumption.

I can quite agree with the statement in item 2 under, "furnace capacities," that the fan will do no good unless there is heat in the furnace. I do not reconcile the statement, however, that "fans should not be depended upon to correct improperly designed or poorly installed gravity circulating systems," for as a matter of fact in several thousand instances to my knowledge, the results have been quite to the contrary.

The warm-air furnace needs anything that will make it better and I think the results of this experiment clearly indicate that the warm-air furnace system is greatly improved by the ue of a fan to force the air circulation. If an optimum

condition means anything at all in warm-air heating, it means taking the heat out of the fuel, putting it into the air, and distributing it in convected form to the livable part of the building where it will produce the greatest comfort.

I am in agreement generally with the data presented in this paper, but I challenge the conclusions drawn for two very definite reasons:

First: Because I think the statements or conclusions formerly referred to, are broader than the data would justify, and

Second: Because if left unchallenged they would tend to hamper the progress of a movement that has already been of a very considerable public benefit.

C. C. HARTPENCE: For five years I have been putting in fan furnace systems and I have had some experiences that I would like to pass on to the members here. When I started out I confined myself to air changes of from 10 to 15 min and quite different from the air changes reported in these tests. Gravity air changes will approximate about 30 min. In the case of average fan circulation, the air changes, as in this case, will run around from 15 to 20 min. In public buildings the usual method of circulation air and getting proper distribution really calls for at least a 15 min air change. I have found the necessity for quicker air change and now have standardized on a 10 min change. Therefore I have installed fans, with ample capacity for rapid air change, which immediately brought into the problem the complications of uniform air distribution and sound. I want to call attention to the possibilities of grief, so to speak, that must be anticipated by the fan manufacturers who are considering this fan circulation system.

The use of fan furnace systems is going to increase materially and practically all manufacturers are preparing to supply the demand. Their two main problems in my opinion are uniform distribution and sound.

Furnaces today both of the cast-iron steel types are not designed primarily for the use of fan distribution. Therefore, the air currents through the furnace are sometimes disturbed and it is very difficult to control even a properly designed system. Faster air flow through a portion of the furnace and irregular heat transfer will create an unbalanced condition in the bonnet of the furnace. This paper emphasizes this extremely important point. Everyone interested in this problem must learn how to distribute the air from the head of the furnace, without objectionable sounds. No doubt furnaces will be specially designed in order to assure predetermined results.

In the layout and installation of a good many systems I have found it easier to predict the results in a large system than in the smaller home plant. I have used every variation of fan furnace system that I could think of, including the use of dust filters, which should be confined to automatic fan jobs with by-pass; I have used humidification and have run into all kinds of problems but sound has been the most troublesome problem. I have tried propeller and centrifugal fans and the outlet velocities I now use are 400 ft with propeller fans and 1300 ft with centrifugal fans. The only way to make a fan furnace job commercially satisfactory and successful is to be able to make it as simple as possible because there are many problems attached.

I believe it is incumbent upon this organization to recognize the advance of fan distribution of heat because of the possibilities of incorporating into that system practically all of the essentials of air conditioning, where engineering knowledge is needed for correct application.

A. P. KRATZ: I think most of the troubles referred to by the first speaker are brought about by a failure to consider that the principles for designing a gravity job are diametrically opposed to the principles for designing a fan job. In gravity work, since the available head depends on the stack height, you must use very large pipes to the first floor in order to obtain the proper amount of heat. On the other hand, in a purely fan job the available head is furnished by the fan independent of the stack height, and the pipes must be installed of a size proportionate to the resistance that is going to be met. The natural result is, that if a fan is installed in a correctly designed gravity job, and that was the proposition we had put before us. The air will take the path of least resistance, which is necessarily to the first floor and the first floor will be overheated. In other words, the system will be unbalanced. We do not question either the feasibility or the advisability of designing a fan job which will operate satisfactorily on a fan, but when such a thing is done, however, the fan must be operated continuously. At the other end of the line there is no question but what a job can be designed and operated satisfactorily on gravity. There is no argument on either of the extreme systems. The argument all comes about from installing a fan in a gravity job or trying to correct a gravity job by means of a fan. The first speaker very well brought out some of those troubles.

In reference to stratification which was mentioned by the second speaker, we could not see any evidence (we made measurements in the different rooms) that the fan had improved the condition so far as stratification was concerned. The temperature differentials from floor to ceiling from measurements made in the different rooms showed no improvement.

A. C. WILLARD AND A. P. KRATZ (WRITTEN): Mr. Miles in his discussion has presented a number of points worthy of consideration, and the authors wish to express their appreciation for the careful study he has given this paper.

We wish to call attention to the fact that no formal conclusions were presented in the paper, and that Mr. Miles' discussion applies solely to the preliminary remarks embodied as an introduction.

Both fans used in the investigation were installed in a correctly designed and installed gravity system, and were operated continuously and not intermittently during the test periods. A correctly installed gravity plant is by no means the remote possibility considered by Mr. Miles. The research work sponsored by the *National Warm Air Heating Association* at the University of Illinois has developed information sufficient to insure the installation of correct gravity plants, and the extensive adoption of the Standard Code for Installation, approved also by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, has given added assurance of the existence of correctly instaled plants.

The objection that the fan system as used was the worst possible because it was installed in the best possible gravity system, is tantamount to an admission that the only excuse for a fan is in a poor gravity system, and therefore, of the correctness of the authors' statement that a properly designed and correctly installed gravity system does not need a fan.

The over-all house efficiency is the only measure, in dollars and cents, of the economic value of any heating system. It is true that the use of a fan in a furnace increases the amount of heat delivered at the register faces for a given combustion rate. This is admittedly a desirable feature. On the other hand, the combustion rate required is determined by the heat loss from the house, and

from the chimney in the shape of flue gases. The fan has no appreciable effect on either one of these, and hence the over-all house efficiency is not increased. In this respect, the fan merely accomplishes a re-distribution of the heat in the house, and not a gain in economy. To offset this, an initiol cost, interest on investment, cost of electrical current, and possibly occasional service costs must be charged against the fan, since these do not exist on a gravity plant. No gain in fuel economy will be accomplished to amortize these charges and there is always the question as to whether the re-distribution of heat is worth the cost.

The suggestion that intermittent operation might be an inherent advantage in utilizing the frequent occurrence of the peaks in heat delivery induced by the fan, is worthy of attention. It is possible that such may be the case, but the authors have no data to either prove or disprove the reasonability of such an assumption, and will have to plead an open mind.

In closing, the authors wish to state that the introductory remarks are generalizations rather than conclusions drawn solely from the material in the paper, and are based on their experience with, and data taken from, the tests on the fan installation in the Research Residence, and on fundamental principles of air flow; and while perfectly willing to concede the right to a different interpretation, they do not see any reason at the present time for changing their own interpretation or the opinions expressed.

No. 864

AIR CONDITIONING THE HALLS OF CONGRESS

By L. L. LEWIS[1] *and* A. E. STACEY, JR.[2], NEWARK, N. J.

MEMBERS

ABOUT 80 years ago the original United States Capitol building measuring approximately 352 ft x 220 ft overall, was found to be inadequate for the purposes for which it was designed. Consequently, in 1851, the construction of the present House and Senate wings was begun. The House wing was completed in 1857 and the Senate wing in 1859.

With these wings the overall dimensions of the building are 751 ft x 350 ft; the building faces east, with Pennsylvania Avenue extending southward from the grounds in the rear. The north wing is occupied by the Senate and the south, by the House of Representatives. It is interesting to note that originally the Senate occupied what is now the Supreme Court chamber, and the House what is now Statuary Hall.

The two chambers are very much alike but of slightly different dimensions. The Hall of the House of Representatives, measured over the Galleries, is 139 ft in length, by 93 ft in width and 36 ft in height from ceiling to floor. In all, 444 people may be seated upon the floor, the outside dimensions of which are 118 ft x 68 ft. Attendants increase this number by 25 or 30. In the galleries there are 616 chairs, 240 people may be seated on the steps and there is standing room for 192, so that on important occasions a total of 1,048 people may be crowded into the galleries.

The Senate Chamber is 113 ft in length, by 80 ft in width and 36 ft in height. The normal seating capacity of the floor is 96, and that of the galleries 682. There are about 20 attendants present. Special sessions may bring all or part of both Houses into a joint meeting, at which galleries and floor may be crowded greatly beyond these limits.

The location of the galleries relative to the floor is shown clearly in Fig. 2. This figure also shows the line of the ceiling, a beamed surface, the panels of which are approximately 8 ft x 10 ft. The center of the panels are frosted and colored glass, in artistic design.

The beams of the ceiling, a false work covering the lower members of the

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roof trusses, are made entirely of cast iron, a most unusual material for such a purpose. The panels do not extend entirely to the side walls, there being a fairly wide border around the outer edges, the material for which, again, is cast iron. A roof of glass covers the attic above this ceiling over practically the entire floor. This is the nearest contact which either chamber has with out-of-doors. The glass roof and the glass ceiling permit the entrance of sufficient light to give a most excellent natural illumination.

Artificial illumination for the entire chamber is now obtained with electric lights, arranged in rows around the edges of the glass panels. These lights may be seen in one of the figures showing the duct work in the attic. (Figs. 4 and 5.) It is interesting to note that at one time this artificial illumination was obtained with some 1,400 gas jets, located in the attic space.

Partly surrounding the floor, and covered by the galleries, are two cloak rooms, the one Republican, the other Democratic. These are "L" shaped, the two longer sides making up the full length of the floor and the shorter sides extending approximately to the center of the shorter side of the floor. While no smoking is permitted on the floor, and, of course, in the galleries, the Congressmen may be completely at ease in the cloak rooms.

Numerous entries in the *Congressional Record* dating back to 1870 show that there has been a continuous and vital interest in the heating and ventilating systems which were first installed when the wings were constructed, and which have been modified from time to time, in order to keep these systems in line with the development of the art, if not well in advance of it.

The scope of the new work was the installation of equipment to air-condition the floor, the chamber, the galleries, the cloak rooms and the press rooms. The systems are designed to cool the conditioned spaces to a temperature of 75 F, with a relative humidity of 40 per cent. In winter the systems are capable of heating to 80 F and coincidentally maintaining a relative humidity of 50 per cent.

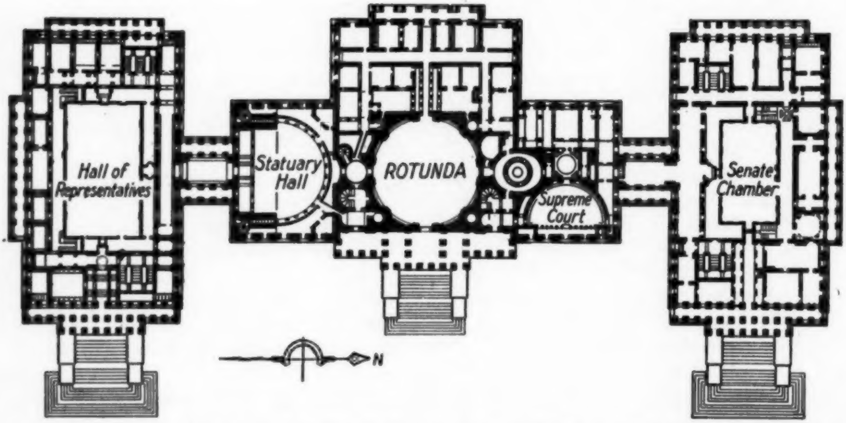

FIG. 1. A DIAGRAMMATIC FLOOR PLAN OF THE CAPITOL SHOWING THE HALL OF THE HOUSE, THE SENATE CHAMBER, THE SUPREME COURT AND STATUARY HALL. THE TERRACES ARE NOT SHOWN

The spaces in which these air conditions are maintained are almost completely shielded from all outside weather variations. Figs. 1 and 2 show that the chambers and cloak rooms are entirely surrounded by the corridors and rooms of the Capitol Building. These enclosing rooms are heated in winter, and in summer will have a temperature approaching that of out-of-doors. The principal duty of the systems conditioning the chambers and cloak rooms is, therefore, to absorb the heat given off by the occupants.

This requirement varies widely and quickly as the people gather first in one place and then another. During an executive session, for instance, the public is excluded and the galleries are empty. During ordinary routine a few tourists are scattered about the galleries; the floor may be sparsely occupied and the cloak rooms crowded with smokers and conferees. A quorum is called for and the cloak rooms are deserted. All of these changes must be promptly met, and counteracted with changes in the cooling power of the supply system.

The installation in the House, which was ready for the short session in December, 1928, and that in the Senate which was first operated in August, 1929, are so very much alike that it seems wise to deal only with the larger of the two installations.

Seven systems are required for complete operation. Of these, five are supply systems, one serving the floor of the House, the second, the galleries, the third, the cloak rooms, the fourth, the attic space, and the fifth, the press rooms. The press room apparatus is independent of the other systems excepting that it receives cold water from the central refrigerating plant. The sixth apparatus is an exhaust system for the removal of air, and the seventh, a forced circulation condenser water cooler for the refrigerating machine.

The problem was so to balance the supply and exhaust that an excess of air would be brought in with the great central stream descending upon the floor and expanded outward to be exhausted through floor openings under the gallery seats. The effect of this is completely to isolate the occupants of the floor from those of the galleries. Thus, one, unfortunate in the possession of a common cold, could enter the gallery to see and to hear and in no way subject a member to the possibility of infection.

A total volume of 36,000 cfm is supplied by the apparatus serving the floor, that is, from system 1. An equal volume is supplied by system 2, serving the galleries. Approximately half of the volume of air handled by system 1 may always be taken from out-of-doors, this coming through a tunnel terminating in a tower intake located about 500 ft from the apparatus in the grounds at the rear of the building. A small amount of recirculation is taken from an exhaust chamber under the floor of the House.

Introducing the Air

The problem of determining a proper method of introducing the air overhead was greatly complicated by three factors; first, that no one had any desire, had it been possible, to change any of the architectural features of the ceiling; second, that grilles or registers were considered too obviously out of harmony with the architecture to be permitted; third, whatever duct work was placed in the attic must not cast any shadows upon the glass ceiling.

Fortunately, many of the decorative features of the ceiling were fastened

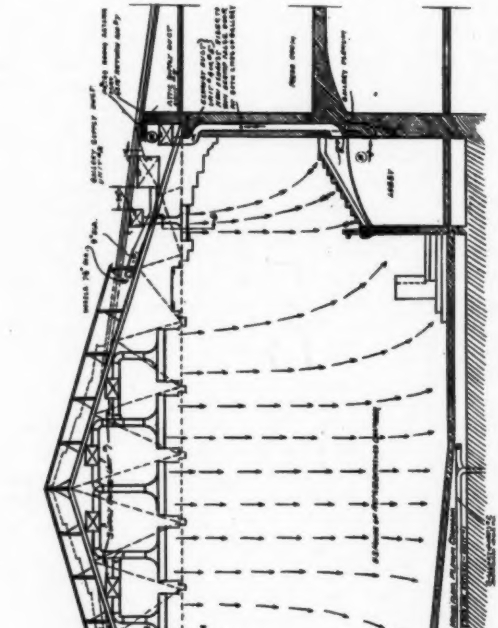

FIG. 2. A TRANSVERSE SECTION. THE SPEAKER ON THE RIGHT FACES TOWARD THE CLOAK ROOMS. OPENINGS FROM THE FLOOR TO THE EXHAUST CHAMBER BENEATH ARE NOT SHOWN

with bolts. The glass panels were framed in heavy wrought iron and these frames could be raised sufficiently to provide space for headers to distribute the air around the edges of the panels. The light and shadow problem was solved by a multiplicity of horizontal and vertical ducts, located as shown in Figs. 4 and 5. Reflection and diffusion of light was increased by painting with a light enamel paint.

Congress is largely made up of members who, having passed beyond the

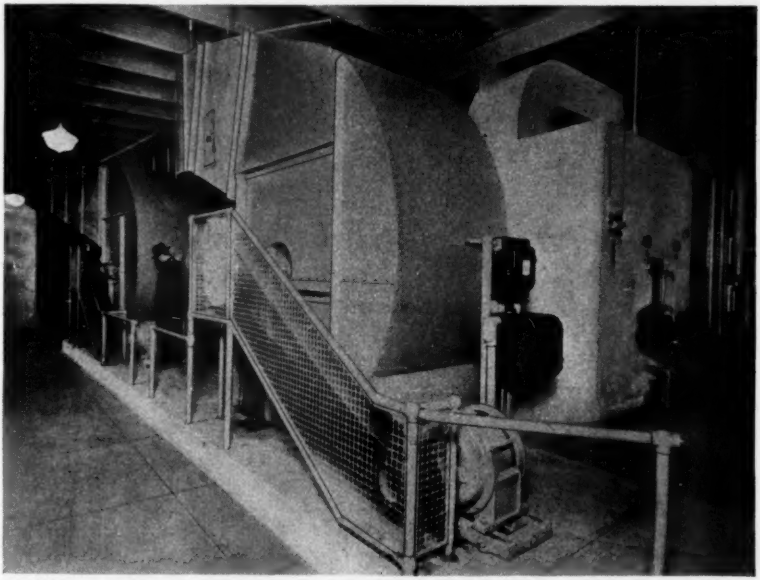

FIG. 3. APPARATUS 1 IS IN THE FOREGROUND. THE MAN LOOKING AT THE DRIVE OF FAN 2 IS FACING TOWARD THE INLET OF FAN 7. THE DUCT RISING DIRECTLY IN FRONT OF THE COLUMN LEADS TO APPARATUS 3

vigor of youth, are sensitive to air circulation. The necessity of making these gentlemen comfortable and the handicaps imposed by architectural features, required that all opinions and theories of the effects of distributing devices, of which Fig. 6 is an example, should be proved prior to installation. Therefore, the important physical features of all questionable sections were duplicated, and diffusers built to various designs and proved experimentally. Figs. 7 and 8 represent section of the ceiling. All tests were conducted with a normal differential of air density for the systems. Observations were made both with smoke and the flames of many candles.

STATIC PRESSURE REGULATOR

The widely varying demands for cooling effect in the area served are met with directly proportional variations in the density of the air delivered from

outlets relative to the density of the air in the chamber. Thus, the tendency of the air to fall bears a direct relation to the cooling work. A slight increase in the velocity of air delivered at maximum density would have a large effect in increasing circulation. An excess is guarded against automatically with a static pressure regulator. This insures a constant supply of air, a constant

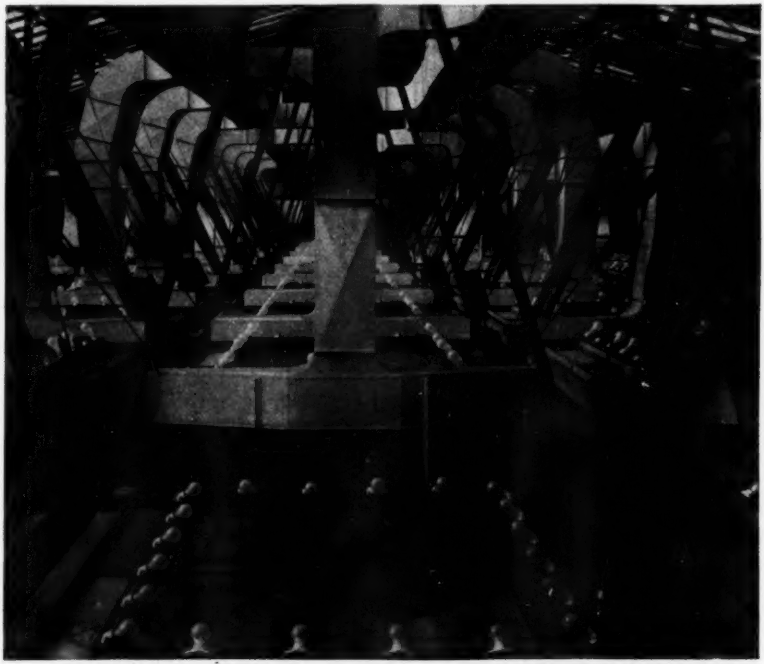

FIG. 4. THE ATTIC FROM A POINT ON THE LONGITUDINAL CENTER LINE. THE COMPRESSION MEMBERS OF THE ROOF TRUSSES ARE CAST IRON

distribution and effectively prevents the air movement in the occupied zones from exceeding a predetermined maximum velocity.

Before placing these systems in commission, careful tests were run on each set of apparatus. As the most tedious and difficult part of the task was balancing the distribution of air at the 90 headers in the ceiling, this was finished first. The static pressures in the header for different capacities had been determined in the laboratory. Volume dampers had been installed in the supply ducts to the headers. These were adjusted to give the proper pressure in each header. A pitot-tube and a sensitive draft gage were used to measure the pressures.

Due to the low air movement in the chambers, dry Kata thermometers were used to determine the velocity. Readings were taken at 32 stations covering

the floor. Flames of candles placed at different points indicated the direction of air movement in the breathing zone. Where necessary, adjustments were made in the openings through the floor to the exhaust chamber beneath. This method effectively balanced the air flow over the chamber. Complete volume tests were carried out on all systems, standard practice being followed.

Fig. 5. Another View of the Attic. The Gallery Outlets and High Velocity Nozzles for the Attic Are in the Background

The resistance thermometers of the 16 point recorder were checked against precision grade glass thermometers graduated to 0.25 F. The recorder was found correct within 0.5 F.

The Ducts

The more important systems are located in the basement room shown on Fig. 9, many feet from the chamber. All of the supply ducts from systems 1, 2 and 3 had to pass through tortuous passages, through heavy stone foundations, and upward through existing flues to the attic space.

It was not possible in every case to run the ducts through the available passages without some modification, and in one case it was necessary to cut through 8 ft of blue stone foundation wall. Two men, assisted by every ordinary means, excepting explosive, spent 18 days enlarging one opening through this wall. In another case it was necessary to utilize the full area of one old smoke flue. The lining of this flue was impregnated with the products of combustion. To remove offensive odors, one and a half courses of the brick lining of this flue were chipped off in order to remove the impregnated portion. The clean surface was then waterproofed, rough plastered and finally covered with a smooth plaster finish.

In order to eliminate soot and smoke as well as dirt, all of the air is passed through oil filters. System 2 supplies 6,000 cfm to the two cloak rooms, a

FIG. 6. ONE OF THE DIFFUSERS FOR THE GALLERY SUPPLY

portion of this being taken from the recirculating passage of system 1. Under maximum load the other portion is conditioned air taken from the dew-point chamber of system 1. The interesting feature of the cloak room system is that all of the air can be delivered either at the floor or ceiling. Part of the air is now being delivered at the floor and part at the ceiling.

System 4 exhausts 18,000 cfm of which 6,000 cfm is removed from ceiling grilles in the cloak rooms and 12,000 cfm from the floor opening in the galleries. The air may be discharged directly out-of-doors, or first used in system 5 to cool the attic space. This 6,000 cfm added to the 12,000 cfm taken from the exhaust chamber under the galleries, makes up the 18,000 cfm total capacity of system 4. This air may then either be blown out-of-doors or into the inlet stream of system 5, the attic supply system.

Attic Supply System

The attic supply system is probably the most interesting of all. It serves the triple purpose of utilizing the air from the chamber to cool the attic space, thereby forming a barrier between outside conditions and the chamber proper. It heats the attic space whenever necessary to prevent the glass ceiling from cooling the upper strata of air in the chamber, and in severe weather it maintains a relative humidity and air movement underlying the glass roof, which will prevent condensation on the surface.

The absolute humidity in the attic might be raised to a point approaching

Fig. 7. Looking Upward Into the Diffuser Slots of a Floor Outlet in the Experimental Set-Up

that of the chamber by leakage through the paneled ceiling, so in winter the inner surface of the roof is swept with a current of warm, dry air which keeps the surface temperature below the dew-point. In summer, the used air taken from exhaust system 4 is delivered through a second set of outlets designed not to sweep the roof but to cool the lower portion of the attic while leaving the hot strata undisturbed in the upper section.

Thus the temperature fluctuations in the attic space are greatly reduced, thereby relieving the automatic regulation of the floor and galleries, reducing the radiation losses from the attic duct work and reducing the temperature head at which air must be delivered.

Temperature Control

The automatic temperature controls installed were designed to meet the unusual conditions previously outlined. For instance, the thermostats con-

trolling the heaters in systems 1 and 2 are located over 200 ft from the apparatus. To prevent the long air lines connecting the thermostat and the diaphragm valves from causing an appreciable lag in the operation of the instruments, some means were required to increase the air capacity of the thermostats. For this reason, air relay valves, which multiplied their effective sensitivity several times, were installed near the thermostats.

A thermostat located in the return air passage controlled the volume of air passing through the dehumidifier. To counteract these changes, a static pres-

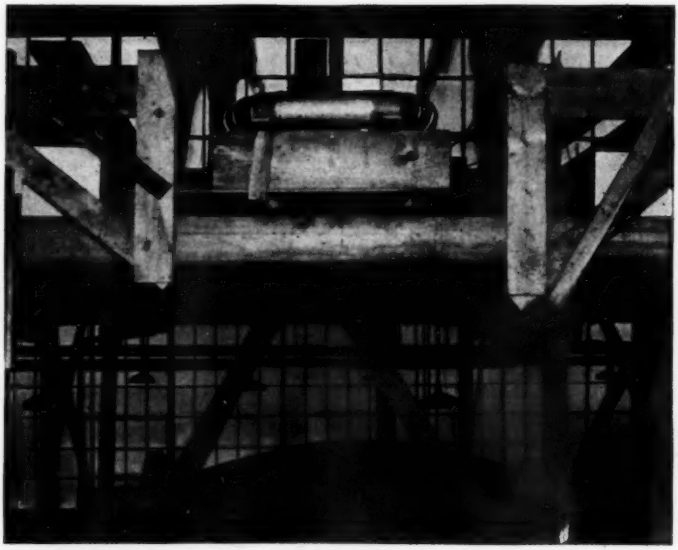

FIG. 8. THE EXPERIMENT WITH SMOKE WHICH IS BEING CARRIED SLIGHTLY TO THE RIGHT BY A GENTLE AIR MOVEMENT IN THE LABORATORY

sure regulator, designed for finely graduated action, alters the position of the recirculating dampers slightly to maintain constant the amount of air discharged by the fan.

Maximum and minimum temperature thermostats are installed in the supply duct, to limit within predetermined ranges the temperature of the air entering the conditioned spaces. These are necessary to maintain the air distribution at the breathing line. A standard dew-point control is used. This is so familiar no description is given.

The sensitive bulbs of the thermostats controlling the tempering coils are located in the outside air duct. As the tempering coils are divided into two parts, an individual thermostat is connected to each section. One thermostat is set to operate at a temperature near the freezing point and the other at 20 F.

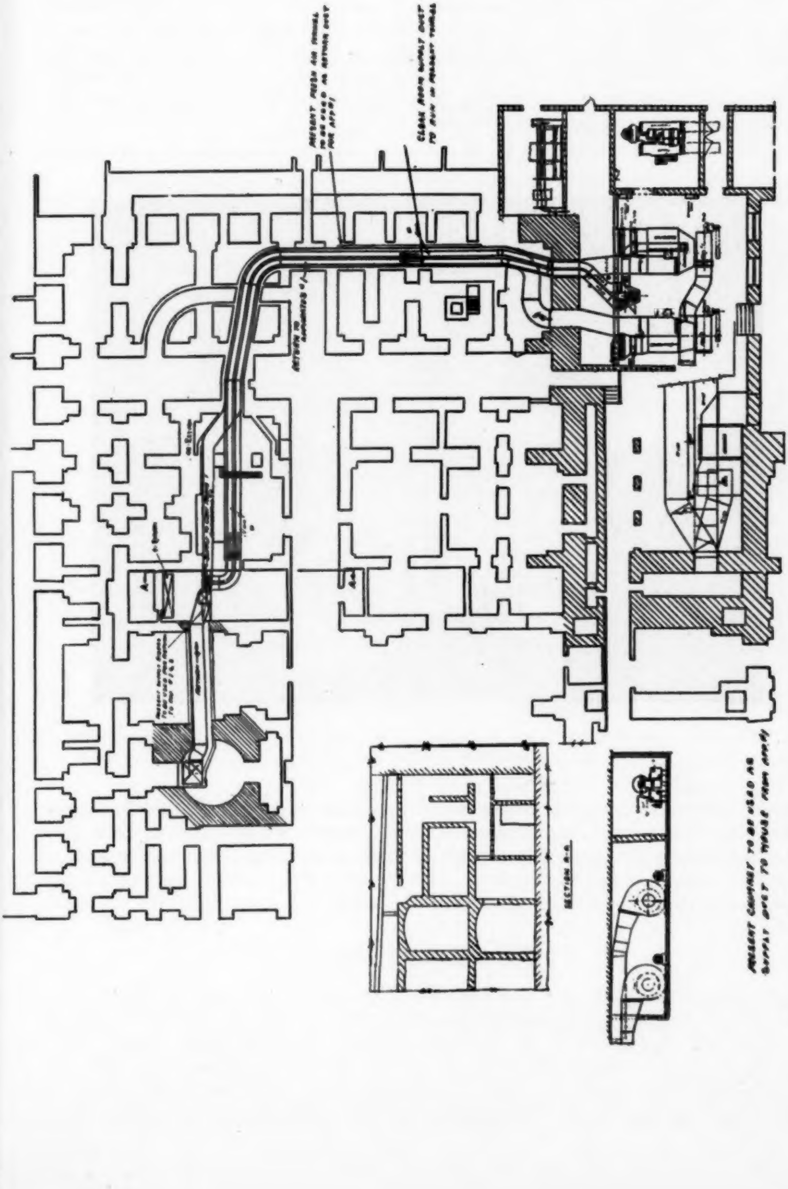

FIG. 9. A PLAN OF THE BASEMENT SHOWING THE GENERAL ARRANGEMENT OF APPARATUS, DUCTS AND REFRIGERATING MACHINERY

Systems 1, 2 and 6 are controlled as previously outlined with the exception that the static regulator was unnecessary on system 6.

The temperature of each cloak room is controlled by a thermostat placed in the room and operating the steam valve on the heater located in the basement. The temperature in the attic space is controlled during the heating period by a thermostat located in the center of the attic, which varies the steam admission to the heaters of system 5. This thermostat is shielded from the direct rays of the sun by white asbestos.

System 6 delivers 7,000 cfm to the five press rooms. The apparatus is located

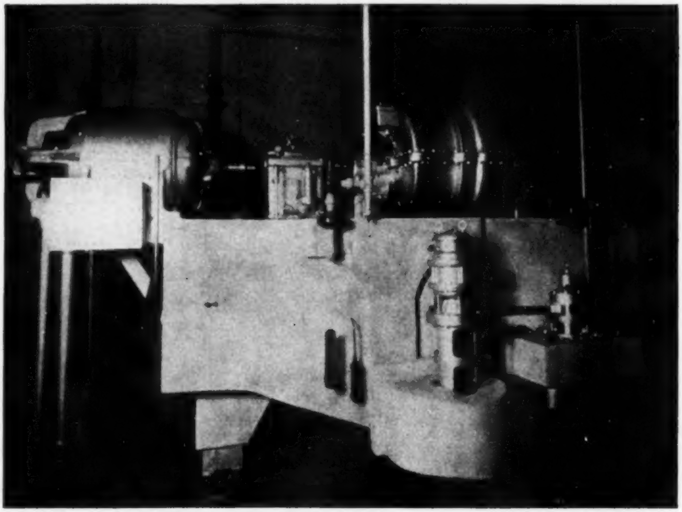

FIG. 10. LOOKING AT THE REFRIGERATING MACHINE THROUGH ONE OF THE OBSERVATION WINDOWS

in the attic near the press rooms and is a replica of systems 1 and 2, except that a static regulator was considered unnecessary. Air is delivered from this apparatus through pan outlets centrally located on the ceiling and spreading cool air uniformly in all directions. Each one of these rooms contains an old fireplace through the flue of which air is relieved.

REFRIGERATING EQUIPMENT

A refrigerating machine, the capacity of which is 206 tons, is placed in a room near the main apparatus room and shown in Fig. 10. It is interesting to note that if all outside air were used instead of partial recirculation there would have been needed 486 tons of refrigeration to maintain the same air conditions in the chamber. The compressor of this machine is driven by an 1,800 rpm alternating current variable speed motor, connected through increas-

ing gears. In order to prevent the noise from the motor and gears from reaching the corridors of the building, this refrigerating machine is entirely enclosed in a room having large windows to permit observation from various points in the apparatus room.

Completely enclosing the refrigerating machine added the complication of removing the heat generated by the motor, radiated from the warmer portions of the condenser, and given off by the resistance grids of the motor controller. Air is admitted through a sound absorber and exhausted from a hood directly above the resistance grids.

Condensing water could not be taken from the city mains, because of an insufficient supply. It was quite impossible to find any location in which either a spray pond or a cooling tower could be hidden on the roof of the building. Such equipment would have been entirely out of harmony with the architecture and purpose of the building. To meet this condition, system 7 was installed. This consists of a fan drawing 49,000 cfm from an interior court through an air washer and discharging it through a vertical flue arising to the roof. The exhaust from the refrigerating machine room is connected to the inlet of the same fan.

In both the condenser and the cooler of the refrigerating machine the water passes through small tubes in a completely closed circuit. The condenser water pump, therefore, withdraws water from the tank of the condenser water cooler of system 7, and forces it through the condenser and back to the sprays, under pressure. A single pump withdraws the used cold water from the tanks of systems 1 and 2, and likewise pumps through the cooler to the sprays. A third pump supplies the necessary volume of cold water to the press room system in the attic. All horizontal runs of water piping are placed in trenches beneath the floor.

For ideal operation, all of the seven systems and the refrigerating machine should have been located within a single room, but this was impossible. With the exception of the press room system the essential features of this plan were put into effect. An operator must ascend to the attic to start and stop the two fans in the attic, but otherwise they can be depended upon to operate without attention.

There are points in the apparatus room from which the refrigerating machine, the condenser water cooler, supply systems 1, 2 and 6, all pumps and the air compressors can be seen. A temperature recorder which automatically prints 16 temperatures upon a broad tape is prominently located on one wall so that the operator may at all times know exactly how the plant is operating.

DISCUSSION

W. H. Carrier: May I emphasize a point that was not clearly brought out? Mention should be made of the old system, installed sometime in the nineties and which served for many years. The air was introduced from a plenum underneath the floor of the House and of the Senate and the system was considered ideal for auditoriums many years ago. In actual operation, however, the air had to be introduced at a temperature not lower than 68 deg because

it was brought in from the plenum through the legs of the chairs. Cooler air would settle along the floor and the senators complained. The system looked to be ideal on paper, the incoming air carrying heat away upward, but when the temperature was lowered complaint of cold was made and if the temperature was raised, it was too hot. Report of the House being stuffy, was not because of inadequate ventilation, for nearly as much air was supplied by the old system as is supplied by the present one. The main difficulty could be traced to the method of introduction and the dust content of the air. The new system has provision for air cleaning by both filtering and washing.

The overhead distribution, permitting lower entering air temperature, has provided comfort both in winter and in summer. The winter ventilation is so greatly improved over what they had before that it was immediately noticeable. A person going from the House into the Senate, both chambers having practically the same air supply, would say that one had good ventilation and the other was very poor. As a matter of fact, the same amount of air, relatively is present in both places, but the same effect is not produced. This is true of all auditoriums and our changed ideas with reference to air introduction is of great importance in maintaining conditions of comfort.

PRESIDENT HARDING: I do not want to let this opportunity pass without thanking the authors for presenting this interesting paper. Of the many problems in heating and ventilating, that of dehumidification is the most complicated and the most difficult to solve. The design of this system, I believe ranks with the ventilating system installed in the Holland Tunnel. The problems encountered in the design of this system were so unusual that I hope at some future time the authors will favor us with another paper covering the calculations involved.

No. 865

TESTS OF DISC AND PROPELLER FANS

By A. I. Brown,[1] Columbus, Ohio

MEMBER

ALONG with the industrial growth of recent years there has developed a greater need for fans for man-cooling and for the removal of fumes and obnoxious gases in industrial plants; greater needs for the cooling of offices, restaurants and kitchens have also been realized. These needs have been filled largely by the increased application of fans of the disc and propeller types, and as a result of the increased demand a number of new designs of fans of these types have been developed. Prominent among the newer designs are the fans of the propeller types having blades similar in shape to the airplane propeller.

In many installations considerable saving in power consumption has been effected by the replacement of centrifugal fans by disc fans, and in other cases opposite results have been found, and many controversies have arisen as to the comparative performances of the various types and designs.

Reliable performance data for centrifugal fans are available but similar data relating to disc and propeller fans are limited, especially in the case of fans of the airplane propeller type.

Objects

The purposes of this paper are two-fold: (1) to note and discuss some observations which the writer has made in the testing of disc and propeller fans, and (2) to present typical data and performance curves for fans of the airplane propeller type.

The observations herein noted are based on commercial tests of fifteen or more fans of various sizes and designs which have been conducted in the Mechanical Engineering Laboratory of the Ohio State University, and the data are presented on subsequent experimental work which is the outgrowth of these commercial tests, and which is now in progress as a project of the Engineering Experiment Station of the same University.

Fan Ratings

The most favorable field of service for disc and propeller fans is in moving

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relatively large quantities of air against small resistances, as in this field they are generally less costly than the centrifugal fans and occupy less space. Certain disc and propeller fans are designed especially for moving small quantities of air against higher resistances, but the majority are found in the former field of service and the manufacturers therefore usually rate their fans on the basis of their free air capacity in cubic feet per minute, and the accompanying speed in revolutions per minute and power input in watts or horsepower.

A little study of these ratings will show some interesting comparisons and some gross errors or exaggerated claims, based, no doubt, upon the results of inaccurate methods of air measurement, and in a few cases with no apparent basis other than that Manufacturer *A* believes his fan to be superior to that of Manufacturer *B* and therefore feels justified in publishing ratings which are somewhat more optimistic.

As an illustration of widely different claims for fans of similar design, the following is noted in the catalog data published by two manufacturers. The first, a large manufacturing concern with an extensive research department, rates their 36 in. fan at 10,700 cfm when running at 945 rpm with an input of 1 hp. The second, a smaller and less widely known concern, rates a smaller fan, their 24 in., at a displacement of 35,000 cfm when running at 1,200 rpm with the same input of 1 hp. If by displacement the manufacturer means the rated air capacity, the capacity of the latter fan is claimed to be 3.27 times that of the former. The first manufacturer further claims for his fan an efficiency of 31 per cent for free discharge. The second manufacturer publishes no values of efficiency but if he had done so on the basis of his published ratings he would have had to claim an efficiency for free discharge of over 3,000 per cent, as shown by the following analysis.

POWER REQUIRED BY A PERFECT FAN

If a fan could be made to operate without any losses, such as those due to eddy currents, air friction and leakage, the power input would be equal to the power required to move the air against the existing pressure. The power required to move the air as expressed in horsepower is equal to the weight of air in pounds per minute, times the head in feet of air, divided by 33,000 ft-lb per minute. This expression when converted to more convenient terms states that the air horsepower equals the volume of air in the cubic feet per minute times the total pressure in inches of water, divided by 6,356. (See A.S.H.V.E. Code for Testing Centrifugal and Disc Fans.) When a fan is discharging freely into the atmosphere no appreciable static pressure exists at the plane of the fan blades, and the total pressure therefore is equal to the velocity pressure, in inches of water, or

$D\left[\frac{V}{1096.2}\right]^2$ where V is the velocity in feet per minute and D is the density of the air in pounds per cubic foot. For average conditions of barometer and temperature the density of the air is approximately 0.072 lb per cubic foot, and by substituting this value for D, the expression for velocity pressure becomes $\left[\frac{V}{4100}\right]^2$.

Applying this expression to the ratings cited for the 24-in. fan, 35,000 cfm

must pass through the opening of the fan, the area of which in this case is approximately 3.5 sq ft, with a resulting average velocity of 10,000 fpm; the velocity pressure, $\left[\frac{V}{4100}\right]^2$ is therefore $\left[\frac{10{,}000}{4100}\right]^2$ or 5.95 in. of water, and the power required to move the air is $\frac{35{,}000 \times 5.95}{6356}$ or 32.7 hp. No account has here been taken of the decrease in the opening of the fan due to the space occupied by the fan blades and shaft, but if such account were taken the horsepower requirement would be even greater than the calculated value of 32.7 hp. The manufacturer, in claiming to perform an amount of work equal to 32.7 hp with a 1 hp motor, is obviously claiming the impossible, and whether unknowingly or by intent, is taking an unfair advantage of his competitors who have honestly and intelligently rated their fans.

TABLE 1. FREE-DISCHARGE CAPACITY OF FANS IN CUBIC FEET PER MINUTE AT 100 PER CENT EFFICIENCY*

Horsepower Input to Fan Blades	Diameter of Fan Opening or Outlet, Inches				
	12	18	24	30	36
0.25	2550	4370	6420	8640	11000
0.50	3210	5500	8100	10900	13860
0.75	3670	6290	9250	12440	15830
1.00	4040	6920	10180	13690	17400
1.50	4620	7930	11650	15660	19950
2.00	5100	8740	12840	17280	22000

* Based upon average air density of 0.072 lb per cubic foot.

If it were possible to make a fan to operate with 100 per cent efficiency, in the case cited the volume of air which could be passed through an opening of 3.5 sq ft by the expenditure of 1 hp would be 10,920 cfm or only 31 per cent of that claimed by the manufacturer. It should be noted that the air capacity claimed by the manufacturer exceeds that of a theoretically perfect fan in the ratio of 3.2 to 1, whereas the horsepower requirement exceeds that of the theoretically perfect fan in the ratio of 32.7 to 1. The horsepower varies as the cube of the fan capacity, and therefore an analysis of performance on the basis of the horsepower required to move the air will magnify discrepancies in ratings that are not so apparent when comparisons are made directly on the volume (cfm) basis.

Table 1 shows the free-discharge capacities of fans of various diameters that would be possible if they could operate without air friction or leakage or any other losses. If a fan actually delivers 75 or 80 per cent of the tabulated value, its performance should be considered excellent.

In comparing the rating of any fan with the values of Table 1 it should be noted that some manufacturers catalog their fans upon the diameter of the blade circle, others upon the diameter of the fan opening, and still others upon the

over-all dimensions. For any values of the diameter of the fan opening other than those shown in the table, at equal power inputs the capacity varies as the $\frac{4}{3}$ power of the diameter ratio; and for a fixed diameter of fan opening the capacity varies as the cube root of the ratio of power inputs.

METHODS OF TESTING

The testing of disc and propeller fans is somewhat more difficult than the testing of centrifugal fans owing to the lack of a satisfactory means of measuring the velocity of a swirling stream of air.

A common but unsatisfactory method of testing consists of traversing the stream of air on the discharge side of the fan with an anemometer or a Pitot tube. Results obtained by this method are inaccurate not only because of the error in measuring a swirling flow but also because of the difficulty in measuring the cross-sectional area of the stream of air which may be greater or less than the area of the opening of the fan and furthermore varying at different distances from the fan. The velocity also varies in different parts of the stream, and with some designs of fans the air near the fan blades may even be flowing in opposite directions, although this fact might not be detected by an anemometer; it can, however, readily be detected by holding a thread in or near the stream and observing the direction which it takes, or by a smoke test. Where a fan which is intended to exhaust air from a room is continually obtaining a part of its supply by leakage from the discharge side, outside of the room, a measurement of the stream on the discharge side is not a true indication of the useful capacity of the fan.

In order to overcome some of these difficulties in air measurement, some fan testers have employed the method of arranging a short length of metal duct on the discharge side of the fan so as to confine the stream of air within an area of known cross-section. This method undoubtedly produces more consistent results than that of attempting to measure the unconfined stream. It is still, however, more or less subject to the errors due to the swirling effect unless this swirl is straightened out by the use of straightening vanes in the duct, and these vanes offer additional resistance of unknown value. The duct may also very greatly affect the capacity of the fan by converting velocity pressure into static pressure and by reducing eddy currents, as will be shown later.

STANDARD CODE METHOD

The Standard Code for the Testing of Centrifugal and Disc Fans which was adopted in 1923 by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers* specifies a method of testing disc and propeller fans which is quite different from the methods just described. This method is fundamentally that of measuring the quantity of air which is discharged by a centrifugal fan into an air chamber maintained at zero pressure when that quantity is therefore just sufficient to replace that which is being exhausted from the chamber by the disc or propeller fan. Under this method the air flow is measured by a Pitot tube which is located on the discharge side of the centrifugal fan where little or no swirl exists and where the stream of air can be measured with accuracy. A diffuser is required in the air chamber so as to guard against short-circuiting of air

from the discharge pipe of the centrifugal fan, through the chamber, to the disc or propeller fan. A draft gage, connected to an impact tube inserted into the chamber, is used to indicate the pressure in the chamber.

This method involves the set-up of a considerable amount of apparatus and for that reason has not been used as generally as some of the more simple but less satisfactory methods. Without doubt, if purchasers of disc and propeller fans demanded fan ratings according to the A.S.H.V.E. Standard Code method, and if manufacturers generally adopted this method of testing, there would be fewer cases of such unfair and often absurd claims as are now frequently found.

COMPARISON OF RATINGS BY DIFFERENT TEST METHODS

No attempt is here made to compare ratings by the Standard Code method with those by the less accurate methods in that the results by the latter methods are too variable. Few manufacturers state in their catalog data the method or methods used in establishing their fan ratings; a small number, however, have followed the practice of publishing ratings of fan capacity based on tests by the more common inaccurate methods and with different ratings for the two sets of conditions, namely, for free air discharge, and for discharge into a short length of duct, usually of a cross-sectional area somewhat greater than that of the fan opening.

If the duct diameter is larger than that of the fan opening, the velocity at the outlet of the duct is necessarily lower than at the fan, and some of the velocity pressure at the fan is converted into static pressure as the air passes through the duct, with resultant gain in capacity.

It is significant that the published ratings for free discharge are invariably greater than those for the somewhat more accurate method of measurement of the discharge into a short length of duct, whereas in all tests which the writer has conducted according to the Standard Code method the results have been the opposite. The principle of conversion of velocity pressure into static pressure by an increase in the cross-sectional area of the stream of air is usually applied to the design of a centrifugal fan, and it seems logical that it can be applied profitably in the installation of a disc or propeller fan. The metal duct on the discharge side of the fan may furthermore decrease air friction by eliminating or reducing eddy currents that exist near the fan at the location where the greatest turbulence takes place, and in some cases the duct may increase the useful capacity of the fan by blocking off the recirculation of air which occurs in certain designs of fans.

The tests herein reported indicate that disc and propeller fans will normally, if not always, have a greater capacity when discharging through a short length of duct of somewhat greater diameter than that of the fan, and that a short duct even of the same diameter as the fan opening will, in the case of an airplane propeller fan, cause an appreciable increase in capacity.

Table 2 shows the capacities of a number of fans for free discharge and for discharge into a short metal duct. The tests from which these results were obtained were conducted according to the A.S.H.V.E. Standard Code method except with reference to the dimensions of the discharge duct. The Standard Code states that "when disc fans are used to blow into a pipe in actual service the performance is different from the performance when mounted in a

TABLE 2. FAN CAPACITIES FOR AIRPLANE PROPELLER TYPE AND THIN PRESSED METAL BLADES

Test No.	1	2	3	4	5	6	7	8	9	10	11	12	13	14
Type of blades	A	A	A	A	A	A	A	A	A	A	A	A	B	B
Number of blades	2	2	2	2	2	3	3	6	6	6	8	8	4	32
Diameter of blade circle, inches	27.15	28.15	28.15	17.85	22.10	26.38	35.62	26.12	35.62	12.05	18.85	20.90	19.00	17.90
Diameter of fan opening, inches	27.40	26.10	26.10	16.50	20.00	26.75	36.50	26.75	36.50	11.00	17.50	19.50	19.50	18.25
Diameter of discharge collar, inches	30	30	26.38	30	30	30	40	30	40	30	30	30	30	30
Length of discharge collar, inches	84	84	40	84	84	84	40	84	40	84	84	84	84	84
Ratio of collar to fan opening diameter	1.10	1.15	1.01	1.82	1.67	1.12	1.10	1.12	1.10	2.73	1.71	1.54	1.54	1.64
Speed for free discharge in revolutions per minute	1745	1694	1710	1964	1700	1765	1750	1750	1730	2044	1704	1150	1154	1190
Tip speed, feet per minute	12400	12500	12600	9190	9830	12200	16330	11970	16200	6450	8400	6300	5740	5580
No-discharge pressure, inches of water	0.255	0.330	0.314	0.200	0.220	0.353	0.725	0.600	1.31	0.236	0.712	0.372	0.472	0.490
Volume of air in cubic feet per minute for free discharge	4770	6230	5970	3025	3970	4225	10970	4930	13580	1530	3380	4150	4450	2590
Volume of air in cubic feet per minute for discharge into collar	6400	7400	6600	3550	5100	5390	14230	6200	17150	1660	4170	4640	5140	2915
Capacity increased due to collar, per cent	34.0	18.5	10.4	17.3	28.4	27.5	29.8	25.6	26.3	8.5	23.3	11.8	15.5	12.5

Note:—*A* denotes blades of airplane propeller type.
B denotes blades of th n pressed metal.

wall, and consequently allowance should be made for the modified condition. Disc fans for blowing purposes are to be tested with the same standard arrangement of apparatus but in addition there must be placed on the discharge side of the fan a collar of the same diameter as the fan opening and 1 diameter long."

The stipulation of the use of a collar of the same diameter as the fan opening precludes the effect of a reduced velocity of the air in the collar. There is therefore no conversion of velocity pressure into static pressure as in the case where the collar or duct is of larger diameter than the fan opening, and the increase in fan capacity due to the addition of the collar is accordingly lessened.

In the tests recorded in Table 2 it will be noted that the diameter of the collar or discharge duct was the same as the diameter of the fan opening in Test 3 only, and in the other tests ranged up to 2.73 times the diameter of the fan opening. The length varied from 30 to 84 in., a variation which probably has no appreciable effect upon the results.

All values of the electric input to the fan motors and the fan efficiencies are purposely omitted from Table 2 inasmuch as some of the fans which were tested were experimental units in the process of development and the publication of any such data applying to them might be misleading and unfair to the manufacturers. Likewise, the values shown for capacities do not necessarily represent the proper ratings for fans of similar size as they are now being manufactured and are shown here only to aid in later discussion. It is to be noted, however, that in all tests the addition of the collar had little effect upon the power required to drive the fan, in some cases causing a slight increase and in other cases a slight decrease in the power consumption.

Referring to Table 2 it will be noted that in all of the fourteen tests the addition of the duct effected an increase in the capacity of the fan, this increase ranging from 8.5 per cent of the free discharge capacity in test 10 to 34 per cent in test 1. Tests 1 to 12, inclusive, are of fans of the airplane propeller type with cast aluminum blades of various pitch angles, in the majority of designs with the blades of such length as to overlap the fan ring, and in other designs with various amounts of clearance between the tips of the blades and the inside of the fan ring. Test 13 is of a fan of thin blade design such as is commonly used in automobile motor cooling, in unit heaters, and in desk fans. Test 14 is of a disc pressure fan with a large number of sheet metal blades, designed primarily for operation at higher pressures. Fan speeds range from 1150 to 2044 rpm, or with tip speeds ranging from 5740 to 16,330 fpm.

It is obvious that too many variable factors are involved in these tests to permit of determining accurately from these limited data the most advantageous relationship between the dimensions of the duct and the fan or any definite laws of performance; however, some interesting observations may be made by comparisons of some of the data.

Tests 2 and 3 are of the same fan blades but with motors of slightly different size and with a slight difference in axial clearance between the blades and the inlet face of the fan ring. These differences are probably responsible for the decrease of 260 cfm or 4.2 per cent in the free discharge capacity in test 3 as compared with test 2. In the same tests where the fan was discharging into a duct or collar the collar used in test 2 was of 15 per cent larger diameter than the fan ring, whereas in test 3 the collar and fan ring were substantially of the

same diameter. The collar in test 2 permitted of considerable conversion of velocity pressure into static pressure and increased the fan capacity by 18.5 per cent, whereas in test 3 no appreciable conversion of velocity pressure into static pressure was possible and still the collar increased the fan capacity by 10.4 per cent.

As additional evidence of the effect of the collar in test 3 the collar was removed without stopping the fan. This procedure resulted in an increase from 0 to 0.068 in. of water in the pressure in the air chamber from which the test fan was exhausting, thereby indicating that without the aid of the collar the fan under test was unable to balance the capacity of the centrifugal fan which was blowing into the air chamber. The removal of the collar was also accompanied by a noticeable increase in the variation of the needle of the watt-meter which was used to measure the input to the fan motor, thereby suggesting the effect of the collar in dampening out turbulence in the flow of air. Similar evidences were observable in tests of some of the other fans.

In test 10 the collar was 2.73 times the diameter of the fan ring, and the effect upon the capacity of the fan is conspicuously small, namely, an increase of only 8.5 per cent, suggesting that the most advantageous ratio of collar diameter to fan diameter has most likely been exceeded. In general the effect of the collar upon fan capacity, in the case of fans of the airplane propeller type, appears to be more marked in the tests of fans having radial clearance between the tips of the blades and the fan ring than in the tests of fans with blades overlapping the ring. This result is to be expected in view of the fact that in free discharge tests of fans with radial clearance some leakage at the tips of the blades was readily detected, and it is reasonable that a collar on the discharge side of the fan would aid in blocking off this back-flow. No doubt the amount of leakage at the tips of the blades is influenced not only by the clearance but also by the design of the blades, and it is possible that by proper design the leakage may be appreciably reduced.

The jet of air which is discharged by fans of the airplane propeller type is in the form of a cylinder extending a considerable distance from the fan, whereas the discharge from fans with the more common forms of thin sheet metal blades generally flares at a wider angle in the form of a cone. It is to be expected therefore in tests of fans of the latter type that a discharge collar would cause a considerable change in the direction of flow, with consequent increase in fluid friction, so that the collar would show less advantage in the matter of increasing the capacity of the fan. The results of tests 13 and 14 are in line with this expectation in that the increases in capacity due to the collar are only 15.5 and 12.5 per cent, even though the collar was of such a diameter as to permit considerable conversion of velocity pressure into static pressure. These results suggest the question as to whether a collar of the same diameter as the fan would show any advantage in connection with a fan from which the discharge flares at a wide angle.

Tests 8 and 9 were on six-blade fans of design and dimensions very similar to the corresponding three-blade fans which were tested in tests 6 and 7. A comparison of the results for the six-blade versus the three-blade fans shows that the former developed a no-discharge pressure which was in one case 70 and in the other case 81 per cent higher than for the similar sized three-blade fans. If the speeds had been identical the percentages would be slightly increased.

In the matter of capacity the two six-blade fans show free-air capacities 16½ and 24 per cent higher than the free-air capacities of the corresponding three-blade fans. These increases in capacity, however, were gained at the expense of considerable increases in power, in one case an increase of 53 per cent and in the other case an increase of 82 per cent.

The marked increase in power which occurred with increase in capacity is in line with the values shown in Table 1 where it is observed that the power consumed by a fan varies as the cube of the capacity, or as otherwise stated, as the cube of the velocity through the fan. If it is desired to move a certain volume of air it can be accomplished with two lower speed fans with one quarter of the power which would be consumed by one higher speed fan of the same size in performing the same service.

Again referring to Table 1 it will be observed that a 30-in. fan, for example, will deliver under free discharge operation approximately as much air with 0.25 hp input as will be delivered by an 18-in. fan of similar design with eight times as much, or 2 hp input. This observation is of prime importance in the selection of a fan for free-discharge service, and it is this feature of relatively large diameters common to the design of airplane propeller fans which accounts in large measure for their low power consumption.

RELIABILITY OF STANDARD CODE METHOD OF TESTING

In the tests herein reported the measured volume of air entering the air chamber was supplied by a fan discharging through a 24-ft length of 30-in. duct, and driven by an electric dynamometer. By varying the speed of the dynamometer and in some cases by restricting the inlet of the fan it was possible to control the air supply with sufficient accuracy as to maintain a substantially constant pressure in the air chamber, with variations never more than 0.005 in. of water above or below normal. As an additional check on the air pressure in some of the tests an anemometer was fitted tightly in the wall of the air chamber and it was observed to show no appreciable flow of air either into or out of the chamber, thereby also giving evidence that the measured quantity of air which was being supplied to the chamber was equal to that which was being removed by the propeller fan.

The magnitude of the error which may be introduced in tests by this method due to a failure to maintain zero pressure in the air chamber was investigated by observing the pressure in the chamber when various measured quantities of air, greater or less than the normal capacity of the propeller fan, were being supplied to the air chamber. The results of this test show that a pressure of 0.01 in. of water above or below atmospheric pressure represented an error in fan capacity of 115 cfm, which was about 2 per cent of the free discharge capacity of the fan under test. With the ordinary inclined-tube draft gage it is possible to read to a fair degree of accuracy a pressure of one-fifth of this magnitude, which in this instance represented an error in measurement of only 23 cfm or 0.4 per cent of the fan capacity, an error which is much less than that which may be expected in air measurement by the Pitot tube.

One of the most conclusive evidences of the reliability of the Standard Test Code method is shown by observing the pressure in an air-tight chamber when a propeller fan is blowing into the chamber and a similar fan of slightly greater or less capacity is exhausting the air from the chamber; then noting that an

equal pressure, but of opposite sign, exists when the fans are interchanged. A test of this sort was occasioned by the statement of Manufacturer *A* that he was confident his fan had a greater capacity than a similar sized fan of Manufacturer *B*, in spite of the results of separate tests which gave Manufacturer *B's* fan a rating of about 1600 cfm above that of Manufacturer *A's*. *A's* fan was set up so as to blow into the air chamber while *B's* fan was exhausting, with the result that a vacuum of 0.067 in. of water was created in the chamber, showing that *A's* fan was not supplying enough air to compensate for that which was being drawn out by *B's*. When the fans were interchanged the pressure in the chamber was found to be 0.065 in. of water, or substantially equal in magnitude to the vacuum which was observed in the former case.

Measurements of Low Velocities by Orifice Meter

In the tests of the various sized fans the same set-up of apparatus was used. This meant that for free discharge or for discharge through an open collar the velocities in the 30-in. measuring duct ranged from 312 to 3500 fpm, although most of the values were above 800 fpm. When the discharge was restricted in blowing tests the velocity in the measuring duct was even lower than 312 fpm, or much too low for satisfactory measurements by a Pitot tube and draft gage. For measurement of the lower velocities it was therefore necessary either to set up a measuring duct of smaller diameter so as to increase the velocities or else to resort to a more sensitive means of measurement.

The drop in pressure through an orifice at the end of the measuring duct was found to serve as a satisfactory means of measurement of the low velocities without in any way interfering with the use of the Pitot tube for measurement of the higher velocities. The orifice was formed by inserting a metal cone in the air-chamber end of the measuring duct. The cone was centered and held rigidly in place in such a position as to restrict the opening an amount sufficient to build up a static pressure of about twenty times the velocity pressure. For any fixed position of the cone the exact ratio of static pressure to velocity pressure was noted when the velocity pressure was sufficiently high as to be determined accurately. It was noted that this ratio remained fairly constant for all velocity pressures of sufficient magnitude to permit accurate readings on the draft gage, varying usually by not more than one per cent from the average value. For velocity pressures which were too low to be read accurately on a draft gage it was therefore assumed that the ratio of static pressure to velocity pressure would remain constant, and the velocity pressure was therefore computed as a definite percentage of the static pressure. Any error in assuming a constant ratio of static pressure to velocity pressure or, in other words, a constant orifice coefficient for various velocities, is undoubtedly less than the usual tolerance in measurement of air flow.

A cone inserted in the end of the measuring duct is of service not only as an orifice meter but also appears to diffuse air in the air chamber in a satisfactory manner and can therefore conveniently replace the diffuser which is required in the standard arrangement of apparatus.

Airplane Propeller Fans

Within the past few years there have been placed on the market a number of makes of propeller fans in which the blades are of cast aluminum, similar in shape to the blades of an airplane propeller. The manufacturers of these fans

contend that in applying the airplane propeller to the propeller fan they have profited by the large amount of study and experimentation which has been involved in the development of efficient types of airplane propellers. They have, however, published very little or no data on the performance of fans of this type aside from their catalog ratings for free delivery nor have they compared the performance characteristics with those of other types of fans.

It is for the purpose of supplying these data that the results of a test of an airplane propeller fan are shown in Table 3 and Fig. 1. Table 3 shows complete results, a portion of which are shown under test 3 of Table 2. The performance curves of Fig. 1 are plotted from the data tabulated in Table 3, and in shape are fairly representative of the performance of all of the airplane fans which were tested, even though the numerical values of efficiency show better performance than was found in the tests of some of the fans. Slight variations in the shape of the performance curves of some of the other fans have

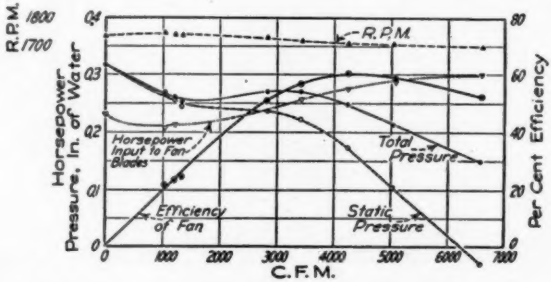

FIG. 1. TEST OF A 2-BLADE AIRPLANE PROPELLER FAN

been found to be due chiefly to the influence of the diameter of the collar into which the fan discharged, and in that respect would not represent results strictly in accordance with the standard test code method.

It may be noted that the power input to the motor, in the data shown, is higher than that required by a number of motor-driven fans of similar size. This is due to low efficiency of the motor and not of the fan, inasmuch as the motor which was used in this test proved to have a maximum efficiency of only 59.6 per cent as against efficiencies of at least 70 per cent found in tests of a number of other makes of motors of similar size.

SUMMARY OF RESULTS

Table 3 and Fig. 1 show, for free discharge, a capacity of 5970 cfm which is 77.8 per cent of the capacity of a perfect fan, and a corresponding fan efficiency of 47.3 per cent. For discharge into a collar of substantially the same diameter of the fan opening the capacity was found to be 6600 cfm, which is 85.4 per cent of the capacity of a perfect fan, and the corresponding fan efficiency was found to be 52.3 per cent.

The maximum fan efficiency of 60.5 per cent occurred when the fan was discharging 4260 cfm against a static pressure of 0.17 in. of water. The power input to the fan blades was a minimum of 0.207 hp at a capacity of 1050 cfm

TABLE 3. TEST OF A 2-BLADE AIRPLANE PROPELLER FAN

Item		Free Discharge	Blowing Tests — Diameter of Discharge Orifice, Inches: Closed	12	14	16	18	20	22	24	Open
1	Center velocity pressure, inches of water	0.097	0.00	0.003	0.004	0.005	0.022	0.032	0.049	0.070	0.118
2	Center velocity, feet per minute	1270	0.00	224	259	289	607	733	906	1080	1405
3	Average velocity in measuring duct, feet per minute	1218	0.00	215	249	277	583	704	870	1036	1348
4	Volume of air in cubic feet per minute = average velocity × area of 4.9 square feet	5970	0.00	1052	1217	1358	2850	3440	4260	5070	6600
5	Calculated velocity through fan ring, feet per minute	1605	......	......	......	......	......	......	......	......	......
6	Velocity pressure at fan ring, inches of water	0.154	......	......	......	......	......	......	......	......	......
7	Static pressure in discharge collar, inches of water	......	0.314	0.264	0.254	0.244	0.236	0.222	0.173	0.105	-0.030
8	Calculated velocity in discharge collar, feet per minute	......	0.00	278	322	359	755	911	1128	1342	1748
9	Velocity pressure in discharge collar, inches of water	......	0.00	0.005	0.006	0.008	0.034	0.049	0.076	0.108	0.182
10	Total pressure at fan ring, inches of water	0.154	......	......	......	......	......	......	......	......	......
11	Total pressure in discharge collar, inches of water	......	0.314	0.269	0.260	0.252	0.270	0.271	0.249	0.213	0.152
12	Horsepower required to move air	0.145	0.00	0.045	0.050	0,054	0.121	0.147	0.167	0.170	0.157
13	Electric input to motor, watts	385	290	263	275	278	300	325	347	365	380
14	Equivalent horsepower input to motor	0.516	0.389	0.353	0.369	0.372	0.402	0.435	0.465	0.489	0.510
15	Speed of fan and motor in revolutions per minute	1710	1732	1743	1738	1737	1728	1718	1710	1707	1700
16	Efficiency of motor, per cent	59.3	59.1	58.6	58.9	58.9	59.2	59.4	59.5	59.6	59.3
17	Horsepower input to fan blades	0.306	0.230	0.207	0.213	0.219	0.238	0.258	0.276	0.291	0.302
18	Overall efficiency of fan and motor, per cent	28.0	0.00	12.6	13.5	14.5	30.1	33.8	35.9	34.7	30.9
19	Efficiency of fan, per cent	47.3	0.00	21.5	23.4	24.6	50.9	56.8	60.5	58.4	52.3
20	Volume of air in cubic feet per minute for 100 per cent fan efficiency	7670	......	......	......	......	......	......	......	......	7725
21	Actual capacity in per cent of that of a perfect fan	77.8	......	......	......	......	......	......	......	......	85.4

Diameter of blade circle, 28.15 in.
Diameter of fan opening, 26.10 in.
Discharge collar, 26.38 in. diam. x 40.0 in.

Corrected barometer, 29.03 in. mercury at 75 F.
Velocity coefficient for center of pipe, 0.96.
Velocity, feet per minute, $4092\sqrt{V.P.}$

and a maximum of 0.306 hp for free discharge. The static and the total pressure curves show a dip in the region of low capacities very similar to that which normally occurs in corresponding performance curves for a multiblade centrifugal fan.

DISCUSSION OF RESULTS

The shape of the power input curve is of particular interest in that it shows that when the discharge is closed the power consumption is considerably less than for free discharge. This feature is opposite to that of the majority of disc and propeller fans, which consume the maximum power when the discharge is closed, and would be even more pronounced if the curves of Fig. 1 had been plotted for a constant speed instead of the actual speed of the motor and fan as it occurred in the tests.

No data are here recorded to show the variations in capacity, pressure, and power consumption which occur with variations in speed, but a number of tests at various speeds verify the expectation that, within the limits of accuracy of measurements, the capacity varies directly as the speed, the pressures as the square of the speed, and the power as the cube of the speed. The highest values of efficiency occur in the neighborhood of the maximum capacity, and it is this characteristic which makes fans of this type particularly suitable for moving relatively large volumes of air against small resistances.

In the case of the discharge through an open collar, the static pressure in the collar was found to be 0.030 in. of water below atmospheric pressure, thereby indicating a contraction of the stream of air within the collar. If this negative pressure had been neglected, on the assumption that zero static pressure existed at the outlet of the collar, the result of fan efficiency would be increased from 52.3 to 62.7 per cent, but this value would not be consistent with the other values of efficiency which were based upon measured pressures inside of the collar.

The efficiency of 47.3 per cent for free discharge is considerably higher than the efficiency claimed for any centrifugal fan for operation under similar service, such values for centrifugal fans usually ranging from 20 to 35 per cent. Similar tests, however, on some of the designs of airplane propeller fans showed efficiencies even lower than the values quoted for centrifugal fans, the variation depending largely upon the amount of leakage at the tip of the blades.

The maximum efficiency which occurred when the fan was discharging against resistance, namely, 60.5 per cent, is not quite as high as the maximum efficiency found in tests of many centrifugal fans, even though it represents much better performance than that which was found in tests of some of the other propeller fans.

In comparing the performance of propeller and of centrifugal fans it should be observed that in the selection of either type for any certain installation a wide range in power consumption is possible, depending upon the velocity at the outlet of the fan, and in order to maintain as low a velocity at the outlet of a centrifugal fan as in the case of a propeller fan, the area of the outlet must be of the same size as the entire opening of the propeller fan, and the entire housing of the centrifugal fan therefore becomes relatively large.

CONCLUSIONS

The claim is often made that the airplane propeller is the most efficient device yet known for the moving of air. If this statement is limited to the free discharge of air or to operation at or near the maximum capacity of the airplane propeller fan it is apparently true. However, the efficiency of a propeller fan is dependent not only upon the design of the propeller itself but to a great extent upon the means for preventing leakage at the tips of the blades.

A collar or short duct on the discharge side of the fan increases the capacity and efficiency, not only by reducing leakage but also by reducing eddy currents, and if the diameter of the collar is somewhat larger than that of the fan the performance is still further improved by the resulting reduction in velocity pressure.

Propeller fans for operation against the higher static pressures have merit with reference to small space requirements and relatively low cost as compared with centrifugal fans but do not show as high efficiencies as are found in tests of centrifugal fans.

Analysis of the fan performance as published by the manufacturers of a considerable number of disc and propeller fans shows unreasonable values of fan capacity and efficiency. The calculated value of fan efficiency and the comparison of the capacity rating with the computed capacity of a perfect fan are serviceable checks on the reliability of catalog data.

The standard method of testing disc and propeller fans as adopted by the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *National Association of Fan Manufacturers* is a reliable method of testing, and shows important features of fan performance which are likely to be overlooked in tests by the older and less accurate methods.

DISCUSSION

H. F. HAGEN (WRITTEN): The author of the paper is to be congratulated for calling attention to the proper method of checking propeller fan ratings. As those of us in the fan industry are well aware, every few years some new and startling propeller fan is introduced, and more often than not the published tables will show over 100 per cent efficiency. Sales can be made on some innovation in design, but frequently the fan disappears from the market.

In connection with the paper, I feel, that a word of caution is needed for the casual reader. To my knowledge, designers of the fan companies have experimented with the stream line section for blades, both on propeller and centrifugal fans. When we learned from the work of the wind tunnel experimenters that they could increase the lift to drag ratio from 8 to 1, which was the maximum they were able to secure with a sheet of uniform thickness, to 18 to 1, we were encouraged to think that we might use this discovery to increase the efficiency of our fans.

The large increase secured by the aeroplane people, and the efficiency which we were already securing on our fans, indicated at once, however, that there was no direct connection between the lift to drag ratio in the aeroplane wing, and the efficiency of a fan. Efficiencies were high enough at the time, so that we knew it would be impossible to multiply them by over two.

My experience has been repeated by that of every other fan designer with whom I have talked on this subject. Greatly to our disappointment we found that the use of the stream line blade section did not increase our efficiencies at all. It seemed permissible, therefore, to infer that the fan action and the lift to drag ratio of the aeroplane wing depended upon different phenomena.

The aeroplane propeller as shown by our experiments is a fan of as high efficiency as the usual type of propeller fan, and no higher. In its two-bladed design the capacity is somewhat too low to meet commercial demands.

If we consider the test curves shown in Fig. 1 of a 28-in. fan at 1,700 with a wide open volume of approximately 6,000, and compare this delivery with that of the average propeller fan as built by the various fan manufacturers this low capacity is strikingly exhibited. The average propeller fan now on the market would in this size and at the test speed, have a free delivery volume according to the standard test code of something over 14,000.

Naturally at commensurate efficiencies, the horsepower taken by the standard fans would be higher. Some of the manufacturers of aeroplane type of fans have stressed this low horsepower taken by their fans without bringing out that the reason for the low horsepower was a still lower air delivery.

There is an extremely attractive idea in applying an aeroplane propeller, with its 80 per cent thrust efficiency, to fan work. However, it should be remembered that thrust is a product of the mass of air handled per second, times the velocity. A good propeller would, perhaps, handle a small mass per second at a high velocity. This would be accomplished by the discharge of air not filling the circle completely, but consisting of two high velocity jets behind each blade.

A good fan action, however, would handle the mass of air filling the whole circle with as slow a velocity as possible, and an ideal fan design would consequently have a minimum of thrust.

This difference is brought out clearly if we consider the tendencies of the two machines. The fan designer has been led to fill in the whole circle with blade area. The aeroplane designer has used two extremely narrow blades. This difference is to be expected when we consider the different purposes for which each is intended.

Considering our own work and the work of others, I don't think that the statement made in the second sentence of the first paragraph under Conclusions is justifiable.

No. 866

SEMI-ANNUAL MEETING, 1930

THE 36th Semi-Annual Meeting of the Society was held at the Curtis Hotel, Minneapolis, June 24 to 27, and was called to order by Pres. L. A. Harding. The excellent program of technical features and entertainment was thoroughly enjoyed by 300 members and guests.

The Program Committee under the able direction of A. C. Willard, *chairman,* R. H. Carpenter and C. F. Eveleth, had provided subjects for discussion which were most interesting, and the attendance at the sessions was a fine tribute to the authors of the technical papers.

E. F. Jones, president of the Minnesota Chapter at that time, introduced the Hon. William F. Kunze, Mayor of Minneapolis, who welcomed the Society members and guests to Minneapolis.

One of the reports presented during the first session was that of the Guide Publication Committee, which was presented by the Chairman D. S. Boyden, as follows:

Report of the Guide Publication Committee

In compiling the ninth edition of THE GUIDE, the Committee is striving to keep pace with the rapid progress of the industry by revising old data to conform to present standards, by adding new material and by eliminating that which is obsolete. It is not the purpose of this Committee to outdo its predecessors, but rather to fulfill its obligation to the Society and to Guide users by making the book as useful and complete as is possible in the light of present available information. Every possible source of information is being used to maintain the high standard of THE GUIDE which has received such wide recognition as the outstanding publication in the heating and ventilating field.

The personnel of THE GUIDE 1931 was completed by May 1, 1930, and about one-third of the revised chapters have already been received. The Committee has been fortunate in enlisting the cooperation of men of outstanding ability for each of the chapters. In practically every case, the associate editors selected are preeminent in the subject for which they are responsible, and each chapter can therefore be relied upon as being an authoritative source of information on that subject.

THE GUIDE 1931 will contain 36 chapters, two of which, Fan Steam Heating and By-Products in Heating, are new. The chapter on By-Products in Heating will include a discussion of Exhaust Steam Heating, and will also include such other data of this nature as are available and appropriate. Some of the material from the paper, Power from Process and Space Heating Steam, by L. A. Harding, p. 53, will be incorporated in this chapter.

Chapter 2 of THE GUIDE 1930 will be divided into four parts in THE GUIDE 1931, three of which will be separate chapters, and the fourth of which will be a part of Chapter 34, Special Applications of Heating and Ventilation. The chapter on Ozone

will be eliminated and a brief discussion of this subject included in Chapter 34, as well as a discussion of Ionization of Air.

The two chapters designated in THE GUIDE 1930 as Air Conditioning and Drying will be treated as correlated chapters in THE GUIDE 1931, and will be designated as Air Conditioning for Comfort, and Air Conditioning for Processing and Drying. In THE GUIDE 1930, the discussion of Unit Air Conditioners and Unit Air Coolers was included in the chapter on Unit Heaters and Air Conditioners. In THE GUIDE 1931, this discussion will be included in the Air Conditioning chapters. Automatic Heat and Humidity Control will be treated in the same chapter in THE GUIDE 1931, although previously these subjects have been treated in separate chapters.

The chapter on Air Ducts has been expanded to include Dampers, Registers and Grilles. Instead of treating Pneumatic Exhaust Systems as in Chapter 28 of THE GUIDE 1930, the subjects of Smoke, Ash and Cinder Treatment will be treated in Chapter 33 of THE GUIDE 1931, as these are considered more closely related to the industry. It is proposed to treat a number of miscellaneous phases and problems relating to heating and ventilation in the Special Applications chapter.

The financial outlook of THE GUIDE 1931 is very favorable. The budgeted income from the Catalog Data Section is $36,500 and already more than 60 per cent of this amount has been contracted for. This is an exceptionally high percentage for this time of the year, particularly in view of the fact that the deadline for copy is still several months away.

The splendid cooperation of manufacturers who have inserted their catalog data in THE GUIDE has helped to increase its usefulness. THE GUIDE is essentially a limited-profit publication and any increase in the amount of catalog data not only increases its usefulness for that reason, but also makes it possible to widen the scope of the material covered in the Text Section. Furthermore, since the difference between income and cost of production goes to the Research Laboratory, any increase in the Catalog Data Section makes possible an increased amount of research investigation, which therefore comes back to the Text Section of THE GUIDE.

It is the intention of the Committee to print 12,000 copies of THE GUIDE 1931.

Respectfully submitted,

D. S. BOYDEN, *Chairman.*

Prof. F. B. Rowley, Minneapolis, Minn., was introduced, and briefly outlined the program of the Committee on Research, and then called upon Director F. C. Houghten to give a detailed report on the various projects.

Report of the Committee on Research

Policies

1. The policies of the Committee on Research must necessarily be governed by the wishes of the Society.

2. In general, these policies have been fairly well established by past procedure. There are, however, specific occasions arising from time to time that require a modification of or the establishment of new policies.

3. *Regulations Governing Committee on Research*

At the Annual Meeting of the Society, a set of regulations amending and bringing up to date all previous regulations governing the activity of the Committee on Research and the Research Laboratory was adopted by the Society. In adopting these regulations, special attention was paid to the question of the Laboratory engaging in commercial research or testing for allied organizations. As a result of this consideration, the permission of the Society for the Laboratory's engaging in such work was re-affirmed and an accepted procedure for undertaking such work was adopted. While commercial research and testing were not made mandatory, the discussion leading up to the adoption of the new regulations emphasized the desir-

ability of the Laboratory engaging in such work under proper conditions and safeguards.

4. One point in which there has always been somewhat of a variation of opinion is in regard to the amount of work which should be done at the Central Laboratory at Pittsburgh in proportion to that which is carried on through various cooperating laboratories. The present policy which seems to have many advantages is to carry on both as needs require without any specific regulation covering the proportion between the two. There is an advantage in the co-operative project in that a broader interest is enlisted in the Society's research work and that many facilities are available both in Laboratory and in trained technical men for carrying out special work which would not be available if confined to one central laboratory. On the other hand, it seems desirable at all times to maintain a Central Laboratory to properly correlate the work, carry on those projects which are not particularly well adapted to co-operative research work, and, also, to uphold the prestige of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS in the research program.

5. Under the new regulations governing the Committee on Research, a request has been received from the *Heating and Piping Contractors National Association* for a program of testing and rating of various types of radiation. Acting upon this request, the Committee on Heat Transmission has outlined a method of test procedure and estimated the approximate cost for the work. This report has been submitted to the *Heating and Piping Contractors National Association*, and as soon as approved by them, the necessary arrangements will be made for doing the work. A few requests have been received for minor tests which do not come under the scope of the Laboratory regulations. In all cases, these have been satisfactorily disposed of by explaining to the applicant the conditions under which certain tests may be made.

Executive Secretary for Research

One of the outstanding changes in the Laboratory organization has been the appointment by the Council of an Executive Secretary to the Committee on Research. The functions of the Secretary are to relieve the Chairman of many of the duties which cause an excessive demand on his time, to place the value of research before the public, to assist in raising funds for research projects in general, to further the interest of the Society in research and to insure the administrative continuity of the Committee.

A man well qualified from past experience and training, Thomas J. Duffield, was appointed February 1 to this position. It necessarily takes some time for a new man to become wholly familiar with all the activities of the Committee on Research and the possibilities connected therewith. It is expected that Mr. Duffield will be able to give a very good account of himself as time goes on, and in order that all may become better acquainted with him and his work, he will present a short report covering the financial aspects of the committee work.

Technical Advisory Committees—1930

Since the field of heating and ventilation covers a wide number of specialties, it is necessary that the Research Laboratory cover a diversified number of projects and carry on research which is of interest and necessity to the various branches of the profession. In order that the Laboratory may be of greater service and in the closest contact with the various interests, each of the major projects which are under investigation are under the general direction of the Technical Advisory Committees. The following are the Technical Advisory Committees which are at present active. New committees are under consideration and will be appointed as new projects are undertaken by the Laboratory.

Air Cleaning Devices: O. W. Armspach, *Chairman;* C. A. Booth, Albert Buenger, Philip Drinker, and H. C. Murphy.

Air Conditions and Their Relation to Health: W. H. Carrier, *Chairman;* A. C. Willard, Philip Drinker, E. V. Hill, W. A. Rowe, and C. P. Yaglou.

Atmospheric Dust and Smoke: A. S. Langsdorf, *Chairman;* E. V. Hill, H. C. Murphy, S. W. Wynne, and O. W. Armspach.

Garage Ventilation: E. K. Campbell, *Chairman;* A. R. Acheson, A. C. Davis, E. B. Langenberg, and W. C. Randall.

Heat Transmission: A. E. Stacey, *Chairman;* A. B. Algren, P. D. Close, A. P. Kratz, and H. J. Schweim.

Oil Burning Devices: L. E. Seeley, *Chairman;* P. E. Fansler, R. V. Frost, H. R. Linn, J. H. McIlvaine, and H. F. Tapp.

Pipe Sizes for Heating Systems: H. M. Hart, *Chairman;* S. E. Dibble, F. E. Giesecke, C. V. Haynes, and R. C. Morgan.

Projects Under Investigation at the Research Laboratory, Pittsburgh, and at the Various Co-operative Institutions

I. Air Conditions and Their Relation to Health

II. Heat Transmission
- (*a*) With reference to built-up wall construction (Univ. of Minn.).
- (*b*) Effect of aging on the conductivity of concrete (Pittsburgh).
- (*c*) Heat absorbed from Solar Radiation (Pittsburgh).
- (*d*) Use of the Nicholls Heat Flow Meter.

III. Infiltration of Air Through Walls
- (*a*) Tests to determine leakage through various types of walls (Univ. of Wisconsin).
- (*b*) A study of the "drift" of air across buildings (Univ. of Wisconsin).

IV. Pipe Sizes for Steam Heating Systems
- (*a*) Capacity of pipes for various parts of a hot water heating system (Texas Agricultural and Mechanical College).
- (*b*) Capacity of pipe for various parts of a steam heating system (by Research Laboratory at Carnegie Institute of Technology).
- (*c*) Study of the use of copper and brass pipe in steam and hot water heating (plans under way).

V. Air Cleaning Devices

A study of methods for determining the amount of dust in air.

VI. Radiation
- (*a*) Plans under way for determining the heat output from various types of radiation (Pittsburgh).
- (*b*) A study of heat distribution from different types of radiation.
- (*c*) Determination of proper method of testing radiators (Purdue).

VII. Garage Ventilation
- (*a*) A study of conditions at Washington University (by Dean A. S. Langsdorf).

VIII. Oil Burning Devices
- (*a*) A study of the operation and method of test (Yale University, Prof. L. E. Seeley).

IX. Atmospheric Dust and Smoke

A study in co-operation with U. S. Weather Bureau and public health department of departments of various cities.

X. Measurement of Air Flow Through Registers and Grilles

(with Armour Institute, Prof. L. E. Davies).

XI. A Study of Thermal Properties of Different Species of Wood

(in co-operation with *National Lumber Manufacturers' Association*).

Co-operative Agreements

The co-operative agreements and contracts are now in force with the following institutions:

Armour Institute of Technology	Texas A. & M. College
Association for Correlating Thermal Research	University of Kentucky
Carnegie Institute of Technology	University of Minnesota
Harvard University	University of Wisconsin
National Lumber Manufacturers' Association	Washington University (St. Louis)
Purdue University	Yale University

Budget

The estimated budget for the year was $36,000. However, due to the fact that the office of Executive Secretary was created after the budget was made out some changes will be necessary.

The Laboratory is supported financially by 40 per cent of the dues from the members and by contributions from associations, manufacturers, and individuals who are interested in the work.

Respectfully submitted,

F. B. ROWLEY, *Chairman.*

The Report of the Research Director was then presented by F. C. Houghten as follows:

Report of the Research Director, June, 1930

SINCE the Annual Meeting of the Society, the Research Laboratory has continued work on the six investigations under way at the Laboratory in Pittsburgh and at the eight cooperating institutions. Five of these investigations were brought to a sufficient state of completion to warrant presentation of technical papers at this meeting as follows:

1. Surface Conductances as Affected by Air Velocity and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw.
2. Capacity of Return Mains for Gravity and Vacuum Steam Heating Systems, by F. C. Houghten and Carl Gutberlet.
3. Loss of Head in Submerged Orifices, by F. E. Giesecke.
4. Air Infiltration Through Various Types of Wood Frame Building Construction, by G. L. Larson, D. W. Nelson, and C. Braatz.
5. Carbon Monoxide Concentration in Garages, by A. S. Langsdorf and R. R. Tucker.

In addition to the investigations resulting in reports presented at this meeting of the Society, the study of heat and moisture loss for men working, and also the study of oil burning devices at Yale University, have progressed sufficiently to warrant preparation of technical reports. However, these were not completed in time to be included in the summer meeting program.

PROBLEMS UNDER INVESTIGATION BY THE RESEARCH LABORATORY IN PITTSBURGH AND IN VARIOUS COOPERATING INSTITUTIONS

I. AIR CONDITIONS AND THEIR RELATION TO HEALTH

Technical Advisory Committee—W. H. Carrier, *Chairman,* A. C. Willard, E. V. Hill, W. A. Rowe, C. P. Yaglou.

Since the Annual Meeting of the Society, the Laboratory has continued the study of heat and moisture dissipated to the atmosphere from the bodies of men working at various rates in still and moving air and also the study of metabolic rates and heat and moisture dissipated from the bodies of children of school age in still air. The series of tests planned for the first mentioned investigation has been completed and a report prepared. Most of the work outlined on the study of heat dissipation from children of school age has been completed and a report of this subject will be available in the near future.

The Technical Advisory Committee on this subject in the past decided that the Laboratory should make a few check tests on the accuracy of the effective temperature lines within the comfort zone and the comfort temperature. These tests have not yet been made but will be as soon as the above studies are completed. The study of vital characteristics of the atmosphere is being continued by Harvard in cooperation with the Research Laboratory.

FIG. 1—DETERMINING THE METABOLIC RATE OF A MAN WORKING IN A STUDY OF THE RATE OF HEAT PRODUCTION IN, AND DISSIPATION FROM, THE BODIES OF WORKING MEN AND THE EFFECT OF ATMOSPHERIC CONDITIONS ON THE DIFFERENTIATION OF THESE LOSSES BETWEEN SENSIBLE AND LATENT HEAT ABSORPTION. MR. TEAGUE OF THE LABORATORY STAFF DETERMINING THE METABOLIC RATE OF MR. HUNTER.

Mr. Carrier's committee has been very active in planning a comprehensive study of the whole subject of relation of atmospheric conditions to the health and comfort of man to be made in cooperation with the U. S. Public Health Service and other organizations interested in this general subject. A tentative plan for this work is under consideration and very satisfactory progress is being made. In connection with these plans a conference attended by Mr. Carrier, Chairman of the Technical Advisory Committee, Dr. Thompson of the U. S. Public Health Service, Mr. Houghten, Director of the Research Laboratory, and Mr. Duffield, Executive Secretary of the Committee on Research, was held at the U. S. Public Health Service in Washington during February and another conference attended by the same persons and also Prof. Drinker of Harvard University, was held in Boston during March.

II. HEAT TRANSMISSION

Technical Advisory Committee—A. E. Stacey, *Chairman*, P. D. Close, A. B. Algren, H. J. Schweim, A. P. Kratz.

1. *Heat Transmission Through Built-Up Walls*—at the University of Minnesota.

Professor Rowley at the University of Minnesota is continuing the study of heat transmission through built-up walls in his guarded hot box. A large number of types of construction have been tested and the results have found their way into the publications of the TRANSACTIONS and THE GUIDE. This work is being continued.

2. *Conductivity of Homogeneous Building and Insulating Materials*—at the University of Minnesota.

The determination of fundamental coefficients of conductivity for various homogeneous building and insulating materials is being continued by the guarded hot plate method at the University of Minnesota. A large number of materials the coefficients of which have been in doubt are being purchased on the open market and tested in order to build up a table of conductivity coefficients of materials on the market at the present time. By agreement with the *Lumber Manufacturers'*

Association a large number of samples of wood are being studied. It is hoped that this table will aid in the preparation of more extensive tables in THE GUIDE.

3. *Study of Surface Coefficients for Various Types of Surfaces for Still and Moving Air*—at the University of Minnesota and Pittsburgh.

Professor Rowley is continuing the study of surface transmission coefficients for various types of surfaces, in the apparatus in his Laboratory designed for that purpose. Coefficients have been published for plain and painted wood, smooth and rough plaster, concrete, stucco, and glass surfaces. Data were also published showing the variation in these coefficients with mean temperature.

A similar study of surface transmission coefficients is being made at the Research Laboratory in Pittsburgh in an apparatus designed by Mr. Harding in cooperation with the Laboratory while he was Chairman of the Technical Advisory Committee on Heat Transmission in 1928 and Chairman of the Committee on Research in 1929. The object of this study is not so much the determination of coefficients for use in practice as it is the study of factors influencing surface heat transfer. In this connection a study was made during the latter part of last year of wind velocity gradients near a surface for a wind blowing parallel to the surface.

Surface transmission coefficients for a painted sand coat finish were determined early this year in the apparatus designed for this purpose. Considerable difficulty was met with in the operation of the apparatus which together with the fact that the coefficient found for this surface was higher than had been used heretofore, resulted in lack of confidence in the findings, for two or three months, during which time a great many check tests and an analysis of the operation of the apparatus were made.

FIG. 2—APPARATUS USED BY PROFESSOR ROWLEY AND HIS ASSISTANTS AT THE UNIVERSITY OF MINNESOTA FOR MEASURING SURFACE TRANSMISSION HEAT FOR STILL AND MOVING AIR

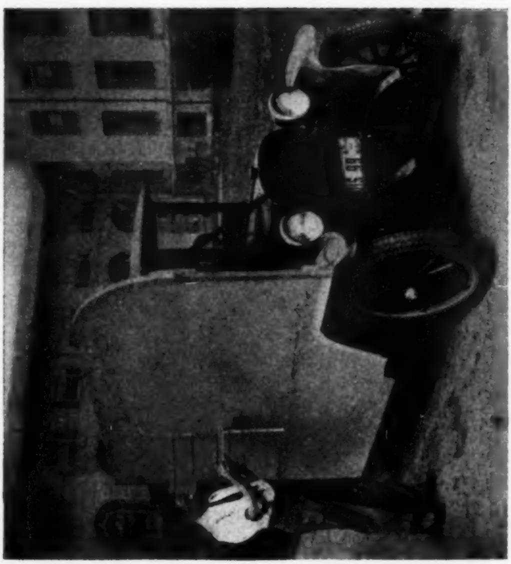

FIG. 3—WALL WITH PITOT TUBES IN PLACE ATTACHED TO A TRUCK AS USED TO DETERMINE THE WIND VELOCITY GRADIENT AWAY FROM A WALL WHOSE SURFACE IS PARALLEL TO THE DIRECTION OF THE WIND. MR. MCDERMOTT OF THE LABORATORY STAFF

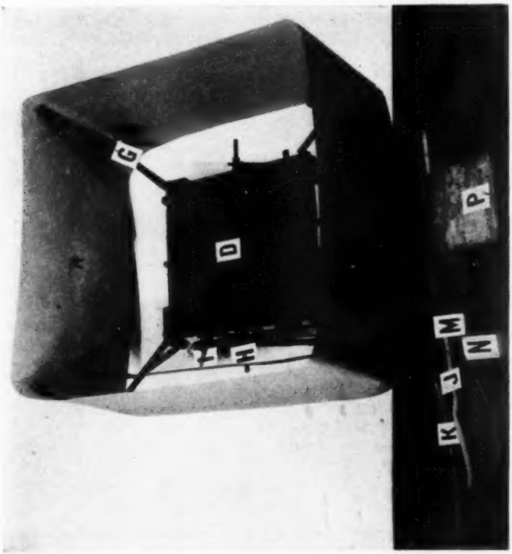

FIG. 4—APPARATUS USED AT THE LABORATORY IN A STUDY OF ABSORPTION OF SOLAR RADIATION IN RELATION TO COLOR AND CHARACTER OF THE SURFACE, ITS ANGLE WITH THE DIRECTION OF THE SUN'S RAYS, AND THE EFFECT OF VARYING AIR TEMPERATURES AND AN INTERVENING PLATE OF GLASS

Recently changes were made in the design of the apparatus after which additional tests were made on a painted, sanded surface and on a smooth, unpainted wood surface.

The change in the design of the apparatus resulted in much more satisfactory operation and in economy in time and the coefficients of the smooth wood surface were found to check very closely similar data by other investigators. Extensive data have been collected showing the air velocity gradients parallel to the surface and the temperature gradients away from it for the apparatus as designed with a 12x12 in. duct and also when duct was reduced in size to 6x12 in. With these data and the data previously collected by the Laboratory on the wind velocity gradient away from a wall it is hoped that important relationships may be worked out. This apparatus was designed with a view of later separating the total heat loss from the surface for given surface and velocity conditions into heat loss by radiation and convection.

4. *Effect of Aging on the Conductivity of Concrete*—at Pittsburgh.

This study is being continued and tests were made during the past six months showing that the present change in conductivity is slow and the study can probably be discontinued, after another year. Tests in the future probably need not be made oftener than once in three months. The present test sample of concrete has been studied for 3 years during the first two of which the conductivity changed very materially verifying the contention of many heating contractors that this is a very important factor in heating a concrete building during the first two seasons.

5. *Heat Absorption from Solar Radiation*—at Pittsburgh.

A paper was presented at the Annual Meeting giving the results of the Laboratory's study of absorption of solar radiation by a wall or roof surface and the effect which angle, color, and temperature of the surface has on such absorption. No plans are now perfected for continuation of the study. However, there appears to be some demand for additional determinations of the effect of solar radiation on heating problems. If additional studies of this nature are to be continued plans should be developed in the near future, so that the work can be undertaken during the summer months.

6. *Nicholls Heat Flow Meter.*

The Laboratory has discontinued its studies of heat transmission through actual building construction under natural weather conditions with the heat flow meters. It is, however, encouraging the use of these instruments for such studies by others and at the present time has lent several of these instruments to the Mellon Institute for use in a study of heat transfer through various types of brick and hollow tile wall construction. Some help in the way of advice and direction for applying the meters and collecting and analyzing the data has been given with a view of assisting in the use of the meters and with a further view of establishing additional data on additional types of walls which may be helpful in the Laboratory's general study of heat transmission. Dr. Anderegg of Mellon Institute is in charge of these studies.

Last year Mr. Nicholls, in cooperation with the Laboratory, made seven new Nicholls Heat Flow Meters. The immediate object in making these meters was to supply the University of Illinois with two for use in their study of a radiator testing room. These two meters were turned over to the University of Illinois early this year. As a result of acquiring the additional five meters the Laboratory has some of these available for distribution to other scientific laboratories interested in research. The Laboratory's object in distributing these meters is two-fold: first, the encouragement of additional research on heat transmission, and second, the expansion of the use of the Nicholls Heat Flow Meter in heat transmission practice.

7. *Cooperation with the National Lumber Manufacturers' Association.*

Professor Rowley and the Research Laboratory have perfected plans for cooperation with the *National Lumber Manufacturers' Association* for the study of heat transmission through and conductivity coefficients of lumber and frame walls. This cooperation will greatly extend the activity of the Laboratory's cooperative program with Minnesota as regards heat transmission.

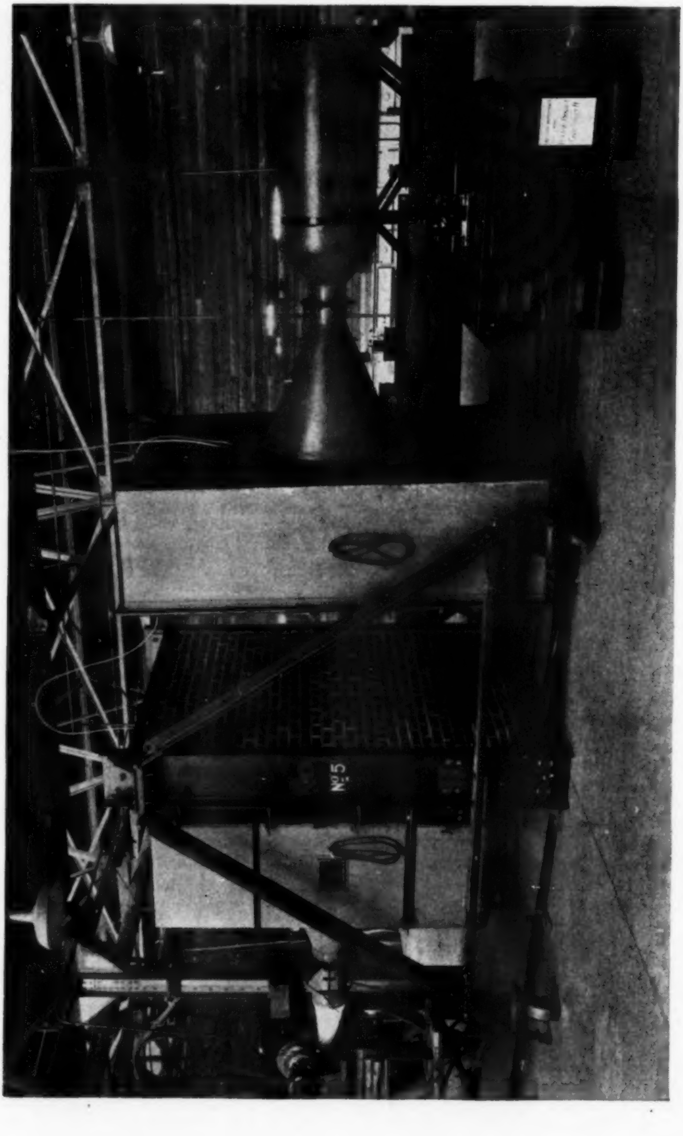

FIG. 5—APPARATUS USED BY PROFESSOR LARSON AND HIS ASSISTANTS AT THE UNIVERSITY OF WISCONSIN IN AN INVESTIGATION OF AIR LEAKAGE THROUGH VARIOUS TYPES OF BUILDING CONSTRUCTION

III. INFILTRATION

Technical Advisory Committee—G. L. Larson, *Chairman*, J. G. Shodron, M. S. Wunderlich, C. C. Schrader, J. E. Emswiler, D. S. Boyden.

Most of the work in this investigation is being carried on by Professor Larson at the University of Wisconsin in cooperation with this Laboratory. A study was recently completed of air leakage through various types of brick walls. These walls are being kept for possible future investigation. The major activity of this study is now on frame walls in cooperation with the *National Lumber Manufacturers' Association*. This work is progressing very satisfactorily and a paper is presented at this meeting by Professor Larson on Infiltration Through Frame Walls.

Periodic tests are being continued on the brick veneer and hollow tile and stucco wall in the two pieces of infiltration apparatus in the Laboratory at Pittsburgh. Results of these two walls should be available for publication during the latter part of this year.

IV. PIPE SIZES FOR HEATING SYSTEMS

Technical Advisory Committee—H. M. Hart, *Chairman*, F. E. Giesecke, R. C. Morgan, C. V. Haynes, S. E. Dibble.

1. *Capacity of Pipe for Various Parts of a Hot Water Heating System*—at the Agricultural and Mechanical College of Texas.

This study has been carried on by Professor Giesecke and his assistants at the Agricultural and Mechanical College of Texas for the past couple of years. A paper was recently given to the Society including a method of calculating pipe sizes for an entire hot water job. This method promises to give the designing engineer a consistent, straight-forward method of determining pipe sizes for hot water jobs similar to the method used for laying out steam heating systems. A great need has been felt for such a series of tables in the past and it is hoped that the results of the studies at Texas will fulfill this need. Plans are now being perfected for continuation or extension of the hot water pipe size study at Texas.

Professor Giesecke is now making a study of the use of orifices in hot water heating systems to insure proper distribution of heat to all radiators in a system. A paper on this subject is being presented at this meeting.

2. *Capacity of Pipe for Various Parts of a Steam Heating System*—by the Research Laboratory at the Carnegie Institute of Technology.

This general subject has been under investigation at the Laboratory in Pittsburgh or at Carnegie Institute of Technology since 1922 and a number of phases of the study have been completed, and the results have been published from time to time. During the past year the Laboratory has been working on the capacity of return risers and dry return mains for vacuum pump and gravity return systems. A large mass of data were collected during the heating season a year ago, which it was hoped would result in a conclusive paper on the subject. However, upon analyzing the data during the summer when steam and working conditions were not available, it was found that data were not sufficient to result in a conclusive paper on the subject and it was necessary to collect additional data under different conditions during the last heating season. As a result of the alterations in Professor Dibble's laboratory, it was necessary to design and build additional features to the laboratory set-up in order to give the additional results desired.

The main difficulties encountered in the study resulted from the fact that such return pipe have almost unlimited capacity provided the system is tight and no additional radiation is turned on from which air must be eliminated through the returns. Whereas, with the addition of comparatively small volumes of air into the system due to leakage in a vacuum pump, job, or due to elimination of air from cold radiators these capacities are greatly reduced. Any conclusions regarding the capacity of return pipe must be based upon the quantity of air handled as well as the radiation supplied. Collection of data was continued during the early part of the present year. A paper, Capacity of Return Mains for Gravity and Vacuum Steam Heating Systems, was

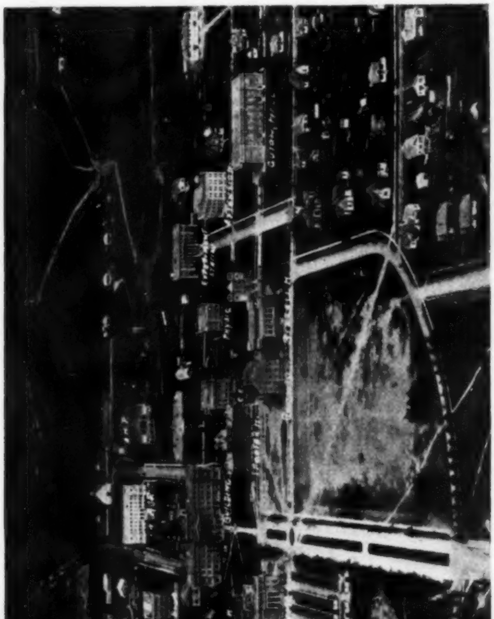

FIG. 6—BIRD'S-EYE VIEW OF THE AGRICULTURAL AND MECHANICAL COLLEGE OF TEXAS, WHERE PROFESSOR GIESECKE IS MAKING A STUDY OF CAPACITY OF PIPE FOR VARIOUS PARTS OF A HOT WATER HEATING SYSTEM

FIG. 7—SET-UP USED BY THE RESEARCH LABORATORY AT CARNEGIE INSTITUTE OF TECHNOLOGY IN A STUDY OF CAPACITY OF RETURN PIPING IN STEAM HEATING SYSTEMS

presented at this meeting. The study is being continued and it is hoped that a paper on the capacity of return risers may be presented to the Society next winter.

3. *Study of the Use of Copper and Brass Pipe in Steam and Hot Water Heating.*

Through the untiring efforts of S. R. Lewis and his interest in the research activities of the Society, plans are being developed for cooperation between the Laboratory and certain copper and brass pipe and tubing manufacturers whereby a study will be made of the comparative carrying capacity of such pipe or tubing in steam and hot water heating systems.

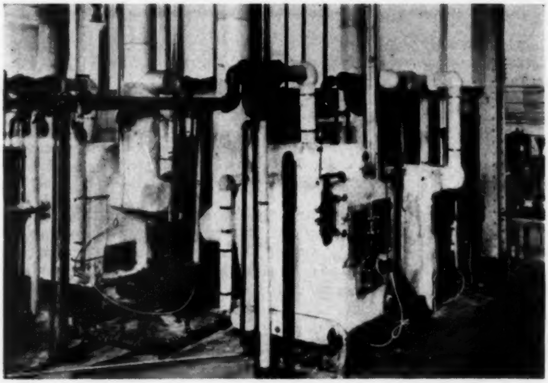

FIG. 8—LABORATORY SET-UP USED BY PROFESSOR SEELEY AT YALE UNIVERSITY IN A STUDY OF METHODS OF TESTING OIL BURNERS IN BOILERS AND FURNACES

V. AIR CLEANING DEVICES

Technical Advisory Committee—O. W. Armspach, *Chairman*, Albert Buenger, H. C. Murphy, Philip Drinker, C. A. Booth.

This study has been carried on during the past few years by Professor Rowley at the University of Minnesota in cooperation with the Research Laboratory. Little progress was made during the past few months, however, due to concentration of effort on the heat transmission studies.

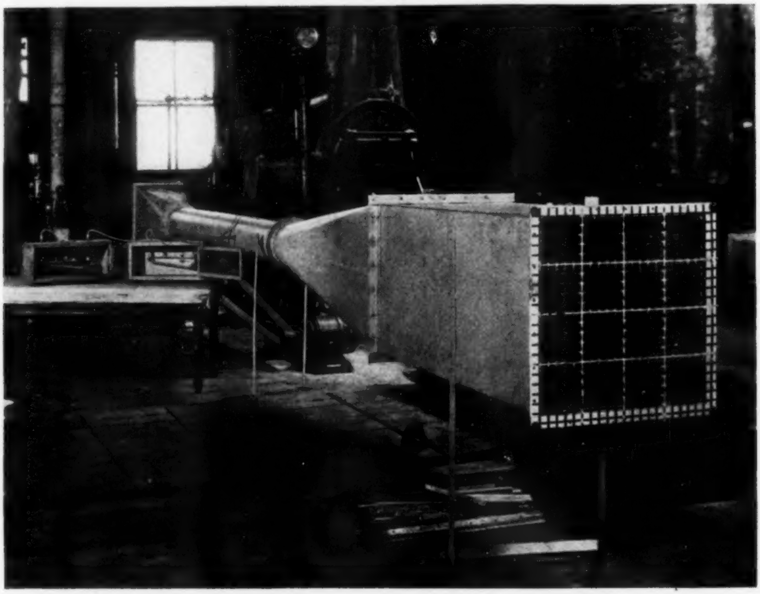

FIG. 9—SET-UP USED BY PROFESSOR DAVIES OF ARMOUR INSTITUTE IN COOPERATION WITH THE RESEARCH LABORATORY AND THE VENTILATING CONTRACTORS EMPLOYERS ASSOCIATION OF CHICAGO IN A STUDY OF METHODS OF MEASURING AIR FLOW THROUGH REGISTERS AND GRILLES

VI. RADIATION

Technical Advisory Committee—A. P. Kratz, *Chairman*, S. R. Lewis, R. S. Franklin, H. F. Hutzel, J. F. McIntire.

The subject of methods of determining the heat emission from various types of radiation has been the most active on the Laboratory's program during the past few months. The main reason for this activity is the desire on the part of the *Heating and Piping Contractors National Association* and others that the Laboratory make heat emission tests on a great many radiators, together with further demand from other groups that the Laboratory make additional studies on the heating effects of radiators in different designs of rooms. The interest in this study has not yet crystallized, however, into a concrete plan.

VII. GARAGE VENTILATION

Technical Advisory Committee—E. K. Campbell, *Chairman*, E. B. Langenberg, W. C. Randall, A. C. Davis, A. R. Acheson.

The subject of garage ventilation was studied during the past winter by Dean Langsdorf of Washington University, St. Louis, in cooperation with the Research Laboratory. The object of the study was to develop information concerning accumulations of CO, gasoline vapors, and other obnoxious gases or vapors in any part of a garage in sufficient concentration to constitute a hazard to health of workers, or a fire or explosive risk. Also to determine the relative effect of various methods and rates of supplying and exhausting air in eliminating these hazards. A number of disappointments were met with in the study during the past year because of the lack of cold weather in St. Louis requiring closed doors in garages. However, considerable data of interest were collected and a paper on the subject was presented by Dean Langsdorf at this meeting. Plans for this study during the next year are now being made by Mr. Campbell's Committee and it is probable that additional studies will be made next winter in a colder climate.

VIII. Oil Burning Devices

Technical Advisory Committee—L. E. Seeley, *Chairman*, H. F. Tapp, P. E. Fansler, H. R. Linn, J. H. McIlvaine, R. V. Frost.

A study of operation of oil burners in heating systems and methods of testing oil burner-boiler and oil burner-furnace combinations is being made by Professor Seeley at Yale in cooperation with the Research Laboratory. This study has been under way for several months and valuable data are now being collected. As a result of these studies a short paper covering a very small phase of the work appeared in the May issue of the Journal and a progress report was given by Professor Seeley at the Chicago meeting of the *American Oil Burner Association*.

IX. Atmospheric Dust and Smoke

Technical Advisory Committee—A. S. Langsdorf, *Chairman*, E. V. Hill, H. C. Murphy, S. W. Wynne, O. W. Armspach.

This study has been dormant for the past few months following the completion of the collection of the first year's data by the U. S. Weather Bureau and the public health departments of the seven cities. Work is now being planned for a study to be made by Dean Langsdorf of Washington University in cooperation with the Research Laboratory.

X. Measurement of Air Flow Through Registers and Grilles

Technical Advisory Committee—S. R. Lewis, *Chairman*, J. J. Haines, John Aeberly, L. E. Davies.

This study has been under investigation by Professor Davies of Armour Institute in cooperation with the *Ventilating Contractors Employers' Association of Chicago* and the Research Laboratory for the past year. A report on the study was made by Professor Davies last winter which gave data on methods of measuring air flow through certain grilles of rather simple construction. Air flow through ornamental grilles is now being studied.

Professor Rowley requested a brief progress report of the finances of the Research Laboratory, which was given by T. J. Duffield.

In a verbal report, T. J. Duffield, the Executive Secretary of the Committee on Research, said that a survey of available sources for funds indicated approximately $42,450 in sight at this time with the possibility of over $45,000, if some anticipated returns are made by organizations, who last year contributed $4,700, but who have made no commitment so far for 1930.

These funds would meet an estimated budget of approximately $40,000 which was an increase of $4,000 over the budget prepared by the Committee on Re-

search, made necessary to meet the expense of travel and secretarial service required in the office of the executive secretary.

It was indicated that more money would be required to meet the program of additional work which has been visualized for the Research Laboratory.

Up to the present time pledges for $8,300 have been received, which is approximately $2,000 under last year's collections, but $3,850 had come in on funds previously pledged. To this might be added that the contribution from the *Heating and Ventilaitng Exposition* in Philadelphia which has been increased from the original $5,000 by $477.

So far the heating and ventilating industry has contributed $13,070 to support the research work which is very fair under existing conditions, but it is hoped that these figures can be raised considerably.

It is a matter of interest that ten organizations have made pledges for more than one year, so that some money can be counted on for next year's work.

The speaker also referred to the contributions made several years ago by the Illinois Chapter to go toward the Research Endowment Fund, so that now $600 is on hand.

In conclusion Mr. Duffield said, "Of course, it had been realized for a long time that we did not want to continue to solicit funds in industry if we could raise money to support our work in any other more dignified way. It was because of that feeling that impetus was given to this movement in the Illinois Chapter. We have counted on the possibility of soliciting funds among some of the philanthropic organizations who always want to aid in good work. I happen to have been associated at one time or another with two of them and I know that they would be sympathetic towards the sort of work we are doing in our Laboratory, but I felt that before we took our pleas to them, it might be a good idea to build up the Research Endowment Fund from within the Society.

"The methods of doing this must come up before the Committee on Research and the Council before anything definite is done, but you should bear in mind that we would like to build the fund up to dignified proportions so that we might then go to the philanthropic organizations and tell them that in addition to the great sum that the Society is contributing toward the support of our current program each year, we have also a Research Endowment Fund of sizable proportions from which we receive income, and we would like to have additions to it."

In commenting on the report, W. T. Jones, Boston, proposed a vote of thanks to C. F. Roth of the International Exposition Co. for the additional contribution to the research fund which resulted from the recent Heating and Ventilating Exposition in Philadelphia. The motion was seconded and unanimously carried.

E. C. Evans, Los Angeles, Cal., said that it was easy to "sell" the Society to Engineers, and pointed out that some addition to our funds would result from the initiation fees and dues paid by recent applicants for membership, who would form the nucleus of a fine chapter on the Pacific Coast.

President Harding said that he had listened to many reports from chairmen of the Committee on Research, and expressed the belief that the present report was the best that had ever been given. He said he did not believe the membership of the Society fully appreciated the value of the research department.

This is, he pointed out, the only engineering organization in the world that maintains its own research department, a committee on research or a research laboratory. When talking about it with other engineers, civil, mechanical and electrical, etc., they seem to be amazed that this organization maintains such a department. Research in reference to the vital characteristics of the air is of the utmost importance to this Society and some other organizations. In conclusion he said, "We delight in giving our renowned scientists praise for the discovery of the so-called structure of the atom and yet to a layman it appears strange that we do not know what are the vital characteristics of the air. To my mind this is the most important piece of research that the Society has before it. It may and will probably take years to finish and will require the cooperation of physicists and physicians. No other organization I believe, has arranged any research program covering the subject. We are on our way. Philip Drinker at Harvard is now studying the ionization of the air. I haven't any doubt that we will find funds forthcoming to carry on the work from other scientific and engineering societies as well as philanthropic organizations."

The report of the Technical Advisory Committee on Oil Burning Devices, of which L. E. Seeley is chairman, was presented by Prof. F. B. Rowley, *chairman,* Committee on Research, as follows:

Report of the Committee on Oil Burning Devices

The chairman wishes to report that since the last meeting of the Society at Philadelphia in January, 1930, much laboratory testing has been done. A test procedure was developed comprising seven types of tests for each combination of boiler and oil burner. This was all explained in a progress report made on April 11 at the meeting of the *American Oil Burner Association* in Chicago. This report was later published in the May issue of the Society's Journal. Four sets of the tests being run were included in the report by way of example.

Since that time more testing of the same nature has been done and the present program is about 60 per cent completed. The work will have to cease during the summer and be resumed in the fall. It is expected that this particular project will reveal operating characteristics of various types of oil burners and boilers.

Incidental to the program of testing is the matter of conducting a test, making measurements, etc. This will eventually call for a test code suitable for a combination of an oil burner and a heating boiler. Considerable work has already been done in this direction and it is the hope of the chairman that the Society will wish to have such a code. The value of the present work being done by the Committee besides all work of this general nature will be enhanced if a standard test code is approved.

Respectfully submitted,

L. E. SEELEY, *Chairman.*

In the absence of H. M. Hart, chairman of the Technical Advisory Committee on Pipe Sizes, F. C. Houghten, Director of the Research Laboratory, gave the following progress report:

Report of Committee on Pipe Sizes

The Research Laboratory has been making a study of pipe sizes for steam and hot water heating systems for the past several years. The work on capacity of pipe for various parts of a steam heating system has been carried on at the Laboratory in Pittsburgh or by the Laboratory staff at the Carnegie Institute of Technology.

From time to time progress on various phases of this study has been presented to the Society, and the results have found their way into the development of the Steam Heating Pipe Size Chapter of the Guide, so that this Chapter of the Guide today, is perhaps, more completely based upon the findings of the Laboratory than any other.

The Laboratory has covered, so far, all phases of the work dealing with the supply side of the system and is now studying return pipe sizes for steam heating systems.

At this time there is being presented a paper resulting from the study of capacities of dry return mains and within the next few months the Laboratory hopes to present its findings on the capacity of return risers which will complete all phases of the study originally outlined.

The Laboratory's work on capacity of pipe for hot water heating systems is being carried on by Professor Giesecke at the Agricultural and Mechanical College of Texas in co-operation with the Laboratory. Satisfactory progress has been made in the development of practical and usable tables for laying out hot water heating systems. However, it is not felt that the ultimate success to be desired in this direction has been attained. In continuing this study Professor Giesecke is now presenting a paper on the resistance of flow through orifices and the application of such orifices to the proper proportioning of a system.

Respectfully submitted,

H. M. HART, *Chairman.*

A report of the Nominating Committee was given by Professor Larson, *chairman,* and the nominees to be voted on at the annual election were as follows:

President, W. H. Carrier, Newark, N. J.

1st Vice-President, F. B. Rowley, Minneapolis, Minn.

2nd Vice-President, W. T. Jones, Boston, Mass.

Treasurer, F. D. Mensing, Philadelphia, Pa.

Members of Council (for 3 year term): Roswell Farnham, Buffalo, N. Y., E. Holt Gurney, Toronto, Ont., E. K. Campbell, Kansas City, Mo., E. O. Eastwood, Seattle, Wash., D. S. Boyden, Boston, Mass. (for unexpired term of Alfred Kellogg, resigned).

W. T. Jones, in expressing his appreciation to the Minnesota Chapter, proposed the following resolutions for consideration:

Resolved, that it is the sense of this meeting that the Committees representing the Minnesota Chapter handling registration, reception, hotel accommodations and entertainment for this Summer Meeting, have anticipated our every want and that we feel highly repaid for making this trip to their beautiful cities.

Resolved, that we extend to the Hostesses of the Minnesota Chapter our sense of appreciation for the graceful and bountiful manner in which our ladies have been entertained.

Resolved, that we extend to the Management of the Curtis Hotel our thanks and appreciation of the excellent service rendered and the pleasant accommodations they have provided for our comfort during our stay in the Twin Cities.

Resolved, that we extend our appreciation for the splendid cooperation received from the technical and daily press in connection with this meeting just terminated.

Resolved, that we express our appreciation to the Chicago, Milwaukee, St. Paul and Pacific Railroad for the wonderful accommodations provided on the Pioneer Limited for our trip from Chicago and for the excellent repasts provided under the masterful direction of George Rector.

President Harding inquired if there was any unfinished business or new business before adjourning the meeting.

PROGRAM 36TH SEMI-ANNUAL MEETING

AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS

CURTIS HOTEL, MINNEAPOLIS, MINN.

JUNE 24-27, 1930

TECHNICAL

Tuesday, June 24

8:30 A.M. Registration, Curtis Hotel

9:30 A.M. Greeting by E. A. Jones, *President* Minnesota Chapter
Response by President L. A. Harding
Report of Committee on Meeting Programs, A. C. Willard, *Chairman*
Control Equipment for Gas Burning Heating Appliances, by W. E. Stark
Economic Use of Steam in Modern Buildings, by F. A. Gunther
Report of Guide Publication Committee, D. S. Boyden, *Chairman*

Wednesday, June 25

9:30 A.M. Report of Committee on Research, F. B. Rowley, *Chairman*
Report of Director of Research Laboratory, F. C. Houghten
Surface Conductances as Affected by Air Velocity and Character of Surface, by F. B. Rowley, A. B. Algren, and J. L. Blackshaw
Wall Surface Temperatures, by A. C. Willard and A. P. Kratz
How Comfort Is Affected by Surface Temperatures and Insulation, by Paul D. Close

Thursday, June 26

9:30 A.M. Report of Committee on Oil Burning Devices, L. E. Seeley, *Chairman*
Report of Committee on Pipe Sizes for Heating Systems, H. M. Hart, *Chairman*
Capacity of Return Mains for Gravity and Vacuum Steam Heating Systems, by F. C. Houghten and Carl Gutberlet
Loss of Head in Submerged Orifices, by F. E. Giesecke
Air Infiltration Through Various Types of Wood Frame Building Construction, by G. L. Larson, D. W. Nelson, and C. Braatz

Friday, June 27

9:30 A.M. The Application of Air Conditioning to Premature Nurseries in Hospitals, by C. P. Yaglou, Philip Drinker and K. D. Blackfan (Presention by W. H. Carrier).
Report of Committee on Garage Ventilation, E. K. Campbell, *Chairman*
Carbon Monoxide Concentartion in Garages, by A. S. Langsdorf and R. R. Tucker
Report of Committee on Air Cleaning Devices, O. W. Armspach, *Chairman*
Report of Nominating Committee

ENTERTAINMENT

Monday

A.M. Headquarters, Reception Committee—LaSalle Hotel, Chicago, Illinois
P.M. Reception of Guests by Committee—Curtis Hotel, Minneapolis, Minn.

Tuesday

A.M. Curtis Hotel
8:30 A.M. Registration—Lobby floor

9:30 A.M.—12:30—Technical Session—Ball Room, Curtis Hotel
11:30 A.M. Luncheon for Ladies
2:00 P.M. Sightseeing and Inspection (Twin Cities and University of Minnesota Engineering Laboratory, Plant of Minneapolis—Honeywell.)
7:00 P.M. Council Dinner and Meeting
8:30 P.M. Informal Reception and Dance, Curtis Hotel Ball Room

Wednesday

9:30 A.M.—12:30—Technical Session—Ball Room, Curtis Hotel
10:30 A.M. Ladies' Shopping Tour
2:00 P.M. Golf Tournament—Minneapolis Automobile Club. (Research Cup)
Ladies' Golf Match—Automobile Club
Bridge and Tea for Ladies—Automobile Club
7:00 P.M. Informal Dinner for Members and Ladies

Thursday

9:30 A.M.—12:30—Technical Session—Ball Room, Curtis Hotel
1:30 P.M. Garden Party for Ladies
2:00 P.M. Golf Meet (Kicker's Handicap)—Thorp Country Club
7:00 P.M. Semi-annual Banquet and Dance—Curtis Hotel Ball Room

Friday

9:30 A.M.—12:30—Technical Session
12:30 P.M. Hostess Luncheon for Ladies

COMMITTEES

General Arrangements Committee: A. Buenger, *Chairman*, H. E. Gerrish, A. J. Huch, F. B. Rowley, H. J. Sperzel, W. F. Uhl, A. M. Wagner

Reception and Registration Committee: F. B. Rowley, *Chairman*. N. D. Adams, W. B. Clarkson, S. A. Challman, G. A. Dahlstrom, Charles Foster, E. B. Gordon, Jr., C. E. Hasey, L. C. Hanson, E. F. Jones, J. V. Martenis, A. H. Probst, A. L. Sanford, M. S. Wunderlich

Hotel and Transportation Committee: W. F. Uhl, *Chairman*, J. H. Brown, E. J. Burns, M. H. Bjerken, H. G. Helstrom, W. N. Parks, F. C. Winterer

Entertainment Committee: H. E. Gerrish, *Chairman*, H. H. Bradford, C. G. Burritt, A. J. Huch, R. B. Mosher

Finance Committee: A. M. Wagner, *Chairman*, E. F. Jones, G. C. Morgan, R. W. Otto

Publicity Committee: A. J. Huch, *Chairman*, C. E. Gausman, C. E. Hill, Fred Shernbeck, E. J. Uhl

Golf Committee: H. J. Sperzel, *Chairman*, D. M. Forfar, J. B. Harris, D. C. Ruff

Ladies and Hostesses Committee: Mrs. F. B. Rowley, *Chairman*

No. 867

APPLICATION OF AIR CONDITIONING TO PREMATURE NURSERIES IN HOSPITALS

By C. P. Yaglou[a] (*MEMBER*), Philip Drinker[b] (*MEMBER*) and K. D. Blackfan[c] (*NON-MEMBER*), Boston, Mass.

DURING the past four years, the authors have had an opportunity to study at the Infants' Hospital (Boston) the growth and development of premature infants under accurately controlled air conditions. The results indicate that mechanical air conditioning can be applied with particular success in the institutional care of premature infants.

The purpose of this paper is to discuss from the engineer's point of view the air conditioning requirements for such nurseries, as these conditions have been determined in this study, and to suggest the types of mechanical equipment suitable for such installations. The medical aspects of the work will be discussed in detail elsewhere.

Nature of the Problem

A premature infant is taken to mean any infant born between the sixth and ninth month of foetal age. These infants are small, weighing between 1½ and 5½ lb, and the majority of them require special treatment if they are to survive. In the same category, so far as hospital care is concerned, are the congenitally diseased and other debilitated infants.[d]

The condition of first importance in the care of such infants is the conservation of body heat and energy, for their vital organs are not completely developed. The metabolism, or internal heat production, is low compared with that of full term, normal infants; moreover, the heat loss is greater than that of normal infants on account of wrinkled skin surfaces and absence of fat deposits in the skin. These facts explain the gradual decline in body temperature after birth, a condition which often persists unless external heat is applied. Inadequate functioning of the heat regulating mechanism in prematures may result in low or in high body temperatures, or, again, in wide fluctuations between the two, depending on the environmental temperature conditions. Wide fluctuations in temperature sometimes prove fatal, since they call on more reserve energy than the premature infants possess. In short, premature infants gen-

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d Congenitally diseased or sick premature infants are those who suffer from congenital anomalies, birth injuries, and acute or chronic infections.

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erally exhibit marked inability to maintain a normal body temperature by their own efforts and they show little resistance to infections, especially respiratory infections. The mortality rate exceeds 50 per cent and has two peaks, one occurring in cold weather and the other in excessively warm and humid weather.

Success in the treatment of premature infants depends mainly on four factors: (*a*) stabilization of body temperature, (*b*) proper nutrition, (*c*) nursing care, and (*d*) prophylaxis against infection. The importance to these infants of adequate ventilation—sufficient air changes without drafts and the maintenance of optimum temperature and humidity conditions—is demonstrable.

PRESENT METHODS OF CONTROLLING BODY TEMPERATURE IN PREMATURES

The usual methods employed for controlling the body temperature of premature infants are the use of heated beds, electric pads, and electric incubators and the application of plenum systems of heating and ventilation. Hess[1] gives an excellent review of the history and development of these methods. The failure to offer a satisfactory solution of the problem of heat regulation in prematures would appear, in the light of the present study, to be largely explained by an insufficient knowledge of the air conditioning requirements and by inability to control the air conditions to the desired degree. In other words, although the importance of air temperature has been generally appreciated, the direct effects of humidity and rate of air change upon body temperature, with their indirect effects on nutrition, growth, infection, etc., have not been fully understood. Most of the methods now in use have failed to fulfill the environmental needs of premature infants on account of limited air space, inadequate ventilation, insufficient humidification in winter, and lack of facilities for cooling and hehumidifying the air in warm weather. Where mechanical ventilation is used, there is a tendency to fix the temperature of a single room at 80 F, regardless of humidity and of the individual requirements of the infants. These mechanical systems, too, are rarely equipped to provide cooling and dehumidification of the air in warm weather. The necessity for such provision is clearly illustrated by the work of Dodd and Wilkinson[2], who discuss the relation between excessive summer heat and obscure fevers in infants.

AIR CONDITIONING REQUIREMENTS IN PREMATURE NURSERIES

From the foregoing discussion, it will be evident to the ventilating engineer that the general requirements for premature nurseries can be met most satisfactorily by the use of a central system of air conditioning which includes cooling as an essential part of the general equipment, even though the latter may be needed for only a month or two each year. Refrigeration is usually available in hospitals at small expense, since most modern establishments are equipped with brine circulating systems.

Humidification of the air in premature nurseries by means of steam humidifiers, hot water pans, or mechanical atomizing devices inside the rooms and dehumidification in warm weather by means of calcium chloride have proved quite ineffectual.

In the air conditioned nurseries of the Infants' Hospital, a ventilation rate of 25 air changes per hour was required to keep the room atmosphere free of odors[e] and to maintain the desired temperature and humidity conditions. An

[1], [2] See Bibliography.

[e] These odors emanate chiefly from loose and frequent stools and vomitus common among prematures.

BLE 1—AIR CONDITIONING REQUIREMENTS OF PREMATURE INFANTS AT INFANTS' HOSPITAL (BOSTON), 1926-1929

Weight Groups Lb	Normal Premature Infants			Congenitally Diseased Infants		
	Room Temperatures, F (Dew-Point, 64 F; Air Movement, 15 fpm)			Incubator Temperatures, F (Relative Humidity of Room, 65%; Air Movement, 15 fpm)		
	Minimum	Maximum	Average	Minimum	Maximum	Average
1.5-1.9	83	88	86.0	86	100	93.5
2.0-2.4	82	87	84.5	86	98	90.5
2.5-2.9	78	86	83.0	84	95	88.0
3.0-3.4	77	86	80.5	79	90	83.5
3.5-3.9	75	85	78.5	79	88	81.0
4.0-4.4	75	85	77.0	78	86	79.0
4.5-4.9	75	85	76.0	76	85	77.0
5.0 and over......	..	80	....	..	85	....

air movement of about 15 fpm over the beds, as measured by the kata-thermometer, proved satisfactory.

It is perhaps unnecessary to point out that neither the comfort zone nor the effective temperature index determined for adults at the Research Laboratory of the Society[3,4] hold for these infants, who differ so much from normal adults in their metabolic and circulatory functions and in the development of their sweat glands.

The preliminary survey showed that the temperature requirements of premature infants vary widely according to their general condition and their body weight. The amount of clothing worn and the humidity of the air are also of importance. Accordingly the infants were classified into two main groups: (*a*) the normal premature and (*b*) the congenitally diseased infants. Each main group was divided into sub-groups according to body weight, as shown in Table 1. The clothing worn consisted in general of a shirt, a diaper, and a flannel jacket; the bedding of two light woolen blankets and a spread. The temperature and humidity requirements for the various groups were determined experimentally over a period of four years (1926 to 1929) by so adjusting the temperature conditions in two adjoining rooms and in electric incubators that fluctuations in body temperature were minimized.

From observations on body temperature, gain in weight, incidence of diarrhea, mortality, and general behavior of the infants under various humidity conditions, a relative humidity of 65 per cent was adopted. When humidities as low as 30 per cent (a considerably higher value than that prevailing in unconditioned nurseries during cold weather) were maintained for two weeks or longer, the body temperature became unstable, gains in weight diminished, the incidence of diarrhea increased, and mortality rose. Continuous exposure to air conditions with 55 to 65 per cent relative humidity, on the other hand, gave very satisfactory results over a two-year period (1928 and 1929). It is of interest to notice that the relative humidity of 65 per cent which was adopted as the optimum for prematures is in close agreement with the humidity conditions recommended by Huntington[5] for adults of the white race.

The temperature requirements for normal prematures (Table 1) varied from a minimum of 75 F to a maximum of 88 F. As will be seen from the same

[3, 4, 5] See Bibliography.

table, the temperatures required by the congenitally diseased group a[illegible]ir various to make group treatment practicable; these infants were therefore [illegible]ry for individually in electric incubators placed inside the larger nursery, w[illegible]e the surrounding air was kept at a relative humidity of 65 per cent. The te[illegible]perature requirements of the normal prematures varied over a range of 13 F, as compared with a range of 24 F for the congenitally diseased infants.

A more thorough discussion of the methods employed in determining the air conditioning requirements and of the physiologic data secured under various air conditions will be published elsewhere.

Table 2 shows how a single central apparatus, furnishing conditioned air at 64 F dew-point to two nurseries, was made to meet the requirements of all the infants. Normal prematures weighing 1.5 to 3.5 lb were cared for in the smaller nursery in an atmosphere of 80 to 85 F and a relative humidity of 50 to 59 per cent. The larger group of normal prematures—those weighing from 3 to 4.5 lb occupied the larger nursery where the temperature was 77 F and the relative humidity 65 per cent. The special requirements of certain infants within each group were met by the use of hot water bottles and by varying the amount of clothing worn. As soon as the infants reached a weight of 5 lb, they were removed to the wards.

On account of the limitations of the air conditioning equipment, the relative humidity in the smaller nursery has necessarily been somewhat below 65 per cent, depending on the temperature of the room. Arrangements are now being made, however, to install auxiliary apparatus which will raise the relative humidity in this nursery also to the 65 per cent desired.

Since the number of congenitally diseased infants to be treated at any one time is small, they can be cared for in individual electric incubators, placed inside the larger nursery with tops open, and the heat adjusted according to requirements. Temperatures as high as 100 F are sometimes temporarily employed. In this way, the infants breathe the comparatively cool room air at about 65 per cent relative humidity and the attending doctors and nurses are not subjected to extreme temperature conditions.

TABLE 2—AIR CONDITIONING SCHEDULE ADOPTED FOR PREMATURE NURSERIES OF INFANTS' HOSPITAL (BOSTON)

Classification of Infants	Weight Groups Lb.	Temperature F	Relative Humidity Per Cent	Air Movement fpm	Method of Treatment
Congenitally diseased and malformed infants	1.5–5.0	(86–100)*	†	15	Infants placed in open top incubators inside large nursery and heat gradually adjusted according to requirements.
Normal prematures	1.5–3.4	80–85	59–50	15	Infants placed in small room and room temperature adjusted according to requirements. Hot water bottles (110F) used in special cases.
	3.5–4.9	77	65	15	Infants placed in large nursery. Hot water bottles (100F) used sparingly in special cases.
	5.0 and over	70–74	Ward	Ward	With exception of definite cases of hypothermia, infants are removed to wards.

* Incubator temperatures taken between basket and metal wall.
† Infants breathe cool room air at about 65 per cent relative humidity.

Since June, 1927, the procedure outlined in Table 2 and described in this paper has been followed with very satisfactory results. Stabilization in body temperature was associated with reduction in the incidence of diarrhea and in the general mortality. Infections were reduced to a minimum and gains in body weight increased. These striking benefits are to be attributed not only to improved air conditions but also to advances in medical treatment and to improvement in nursing and general care. It is fair to conclude, however, that the introduction of mechanical air conditioning has proved a distinct advance in the care and treatment of premature infants.

THE AIR CONDITIONED NURSERIES AT THE INFANTS' HOSPITAL (BOSTON)

General Specifications and Recommendations for Future Installations: The nurseries consist of two adjoining rooms 12 ft x 8 ft x 12 ft and 12 ft x 12 ft x 12 ft, the last dimension being the height. Both rooms are equipped with double windows. At the time the air conditioning system was installed, the larger room was partitioned into cubicles and the partitions were allowed to stand, since they did not interfere with ventilation.

The system was originally designed for 45 air changes per hour in each room, with provision for reducing the air changes to 20 per hour by means of variable speed fan-motors. Later it was found that the actual requirements were only 25 changes per hour, so that it was possible to add a third room to the system thus making use of the full capacity of the air conditioning apparatus. This new room, measuring 11 ft x 20 ft x 9 ft is used as a special observation nursery for full term infants suffering from such ailments as fever, diarrhea, and other gastro-intestinal disturbances.

The apparatus has sufficient capacity to maintain the three rooms at a maximum temperature of 110 F with a maximum dew-point of 80 F, when all the air is taken from out of doors at 0 F. In summer, the rooms can be cooled to 75 F and the dew-point lowered to 60 F when the outdoor dry-bulb does not exceed 95 F and the wet-bulb 75 F.

By referring to Table 2, it can be seen that the maximum winter conditions for which the apparatus has been designed are needlessly high, and that it would be adequate to design such systems in the future for temperature of 90 F and 80 F in the smaller and larger rooms respectively, and for a dew-point of 70 F. The refrigeration capacity of the equipment happens to be quite close to the actual requirements, as shown in Table 2. No recirculation allowance should be made in the design of such systems.

Description of Apparatus: The installation at the Infants' Hospital consists of an individual supply system for furnishing conditioned air to the three nurseries and of an individual exhaust system for withdrawing air from the rooms. The conditioning apparatus is located in the basement.

The supply system consists of the following:

(*a*) Outdoor air intake, containing a weather-proof hood and a screen.

(*b*) Light weight copper tempering coils, controlled thermostatically.

(*c*) A dehumidifier of the horizontal mist spray type for humidifying the air in winter and for cooling and dehumidifying the air in summer. Humidification in winter is accomplished by means of a closed type water heater which warms the spray water. In summer the spray water

is cooled in a Baudelot cooler, on top of which the dehumidifier is mounted. A centrifugal pump circulates the spray water.

(*d*) A supply fan, which draws the air through the dehumidifier and forces it through the rooms.

(*e*) Individual light weight copper reheaters, controlled thermostatically, for warming the air to the temperature desired at the room inlet registers.

(*f*) Individual supply ducts, fitted with dampers and registers, near the ceiling of the rooms for even distribution of the air inside the room.

The exhaust system consists of the following:

(*a*) Individual exhaust ducts, fitted with dampers and registers, near the floor of the rooms for withdrawing the used air from the rooms.

(*b*) Exhaust fan to which the individual exhaust ducts from the rooms are piped.

(*c*) Recirculating ducts for recirculating part of the air at the inlet side of the dehumidifier.

(*d*) Main exhaust outlet duct which discharges the used air out of doors.

The Central Apparatus: The dehumidifier is of standard construction. It is made of galvanized iron and has the following dimensions: width and height, 2 ft 0 in.; length, 9 ft 0 in.; over-all height, including tanks, 3 ft 0 in. A spray system of 27 sprays is arranged in three banks, two of which spray in the direction of the air flow and the third against it.

The water circulating system consists of a centrifugal pump having a rated capacity of 40 gpm at 20-lb pressure at the pump discharge.

The water heater is of the closed type. It is capable of warming 40 gal of water per minute through a maximum of 10 F with steam at 5 lb pressure. Originally the water heater was equipped with a water by-pass and a three-way diaphragm valve for the control of humidity by the wet-bulb temperature method, but this arrangement proved to be unsatisfactory for our purpose and is no longer used.

The Baudelot cooler is 2 ft wide by 4 ft high by 15 ft long. The cooling coils are made up of 2 in. wrought iron pipes, black on the inside and galvanized on the outside. They are arranged in two vertical rows, each 8 pipes high and about 10 ft long. The flooding troughs, made of galvanized iron, are of the saw-tooth adjustable type and extend over the entire length of the coils. The refrigerant is brine at about 12 F, which is taken from the main brine circulating system of the hospital. Under our operating conditions, the refrigeration capacity is about 7 tons.

The fans are of the multivane type, direct connected to variable speed d-c motors, by means of which their speed may be varied between 600 and 300 rpm. The capacity of the supply fan at 600 rpm is about 2,000 cfm when it operates against a static head of about 0.7 in. of water. The capacity of the exhaust fan at the same speed is about 1,500 cfm at 0.15 in. static pressure. According to the authors' experience, both supply and exhaust fans should have the same capacity and should not be direct connected on extended motor shafts.

Fig. 1 shows a general view of the central air conditioning apparatus.

System of Duct Work: The supply ducts extend across the entire length of

the rooms on an interior wall, at a height of 8.5 ft from the floor. Air distribution is secured by means of adjustable louvre registers, spaced along the face of the ducts about 2 ft on centers. The conditioned air is blown across the rooms directly against the exposed walls and glass, which constitute the cooling surfaces in winter and the warming surfaces in summer.

After circulating across the rooms, the air is withdrawn near the floor by means of exhaust ducts, which are similar in all respects to the supply ducts. Figs. 2 and 3 show the general arrangement of the supply and exhaust duct

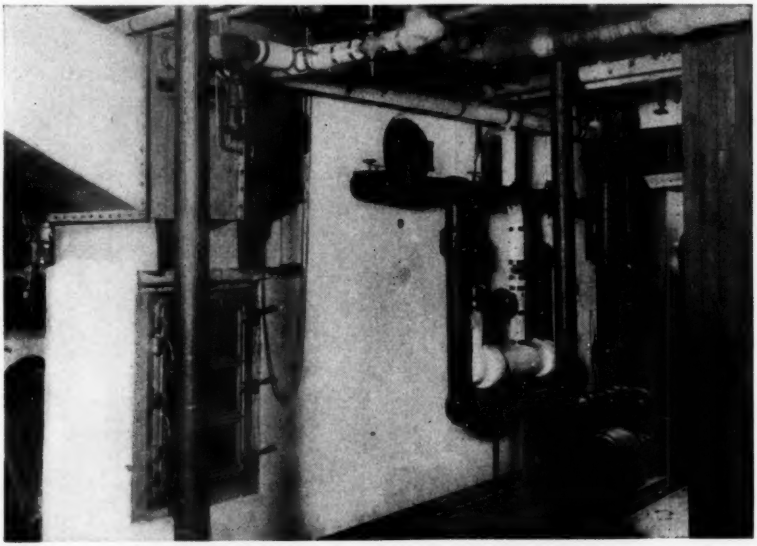

FIG. 1—CENTRAL AIR CONDITIONING APPARATUS

system in the two premature nurseries. They also show the cubicles, beds, supply closet, etc.

Some difficulty was experienced in distributing the air uniformly inside the rooms so as to secure an air movement of about 15 fpm, as measured by the kata-thermometer, in the zone occupied by the infants. This was finally accomplished by reducing the air changes from 45 to 25 an hour and by adjusting the louvres of the registers so as to deflect the air toward the ceiling and thence against the windows.

The reduction in the number of air changes from 45 to 25 an hour has considerably diminished the air noise in the ducts. According to the authors' experience, both supply and exhaust risers for such systems should be designed for air velocities of 450 fpm or less, according to the size of the ducts.

Temperature Control: The problem of temperature control has been a most difficult one, because premature infants seem to be more sensitive to temper-

ature changes than are the majority of the regulators which were tested. The small size of the rooms adds to the difficulty.

In addition to the size of the room and the type of regulator used, accuracy in temperature control was found to depend upon location of thermostats, velocity of air over the thermostats, rate of air change in rooms, type and amount

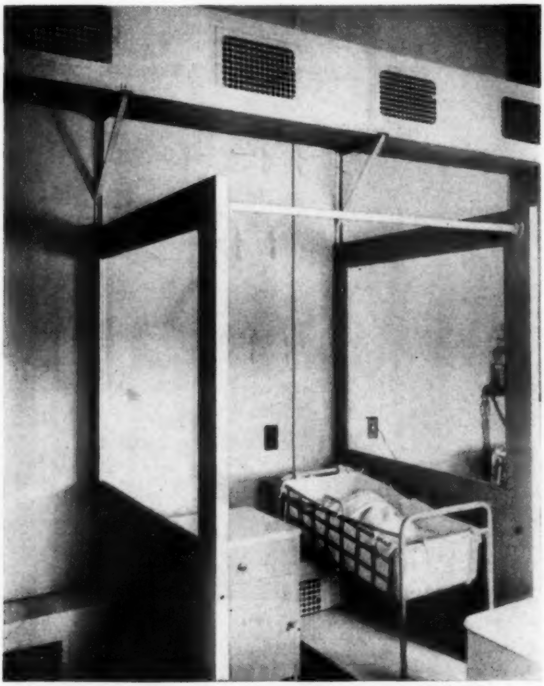

FIG. 2—SUPPLY AND EXHAUST DUCTS IN LARGER PREMATURE NURSERY

of heating surface controlled, type and size of controlled valve, and weather conditions. Some of the results secured with different regulators have been published elsewhere.[6] According to these data, sling psychrometer readings showed that the temperature fluctuations in the rooms ranged from a minimum of 2.9 F to a maximum of 8.9 F, depending upon the type and location of the thermostat.

The best regulation was secured by means of a gas filled capillary coil bulb, made of fine copper tubing 60 ft long, which limited the fluctuation to 2.9 F. After about a year's operation, however, corrosion inside the copper tubing stopped up the capillaries of the instruments and it was necessary to replace

[6] See Bibliography.

them with ¼-in. copper bulbs about 30 in. long. These bulbs are not as sensitive as those of the capillary type, but they are much more dependable. They are exposed inside the exhaust ducts in the region of maximum velocity, where they work best. The operating mechanisms of the thermostats are mounted on the wall just outside the nurseries and connect with the bulbs by means of armored flexible tubing. These thermostats control the steam diaphragm valves on the reheaters by the medium of compressed air. With this arrangement,

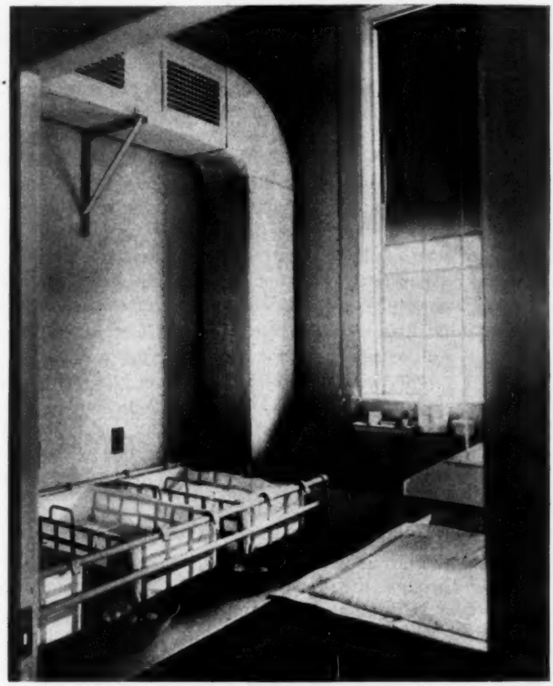

Fig. 3—Supply Duct in Smaller Premature Nursery

it is possible to control the room temperature within a range of 3.5 F, as recorded by the sling psychrometer.

In a more recent installation the temperature fluctuation was reduced to about 2.5 F by the judicious choice of regulators and diaphragm valves.

Humidity Control: The control of humidity by the wet-bulb temperature method was unsatisfactory for this particular installation, because the wet-bulb temperature in the rooms varied as much as 6 F and the system was loaded with many automatic devices, which required constant attention.

By rearranging the regulators and valves and by eliminating superfluous pieces of apparatus, the authors were able to use a central dew-point control,

which proved much superior to the former system. In this modified system, the dew-point regulator is placed in the path of the air leaving the dehumidifier. In winter the regulator controls the supply of steam to the water heater by means of compressed air and thus warms the spray water as desired. In summer the regulator controls a three-way valve on the suction side of the water pump. One side of this valve connects to the pump itself, another side to the refrigerated water in the Baudelot cooler tank, and the third side to the warm water in the dehumidifier tank. The desired dew-point is maintained automatically by mixing the right proportion of cold water from the cooler tank with warm water from the dehumidifier tank. When the dew-point of the air leaving the air washer drops slightly, the regulator partially closes the cold water port of the three-way valve and opens the warm water port sufficiently to raise the temperature of the spray water to the required degree. If the water in the dehumidifier tank is not warm enough to secure the dew-point for which the regulator is set (under this condition, the warm water port is wide open and the cold water fully closed) the steam diaphragm valve on the water heater is automatically opened to raise the temperature of the spray water. The tension of the springs in the three-way valve and the heater valve is adjusted in such a way that little attention to hand valves is required.

The water in the cooler tank is automatically maintained at about 59 F by means of a water thermostat controlling the brine valve to the cooling coils.

Dry- and Wet-Bulb Temperature Recorders: Continuous and permanent records of the dry- and wet-bulb temperatures of the air in the rooms are secured by means of automatic recorders of the remote type, which are mounted on the wall just outside the nurseries. The thermostatic bulbs of these instruments are exposed inside the exhaust ducts in the current of air leaving the rooms where their sensitivity is increased considerably by the air movement. It was found that the condition of the air in this region represented quite closely the average condition of the air in the rooms.

OPERATION OF APPARATUS

In winter the incoming air is warmed automatically by the preheater to about 65 F. In passing through the spray chamber, the air absorbs a sufficient amount of moisture to saturate it at a temperature somewhat higher than the dew-point of 64 F at which the rooms are kept. The difference between the dew-point at the apparatus and that in the rooms represents the loss of moisture by absorption and diffusion through the walls of the rooms. In order to compensate for this loss, the dew-point of the air leaving the dehumidifier is increased systematically 1 F each month beginning in October, until it reaches a maximum value of 68 F in January, the coldest month in Boston. Beginning in February, the dew-point is gradually reduced 1 F each month until May, when it is brought back to 64 F. From May to October, the dew-point in the rooms is practically the same as that at the apparatus, except during short periods of extreme weather.

During the heating season, the reheaters warm the air to whatever temperature is demanded by the room thermostats. Since no direct radiation is used in the rooms, the ventilating current must furnish sufficient heat to balance all the heat losses and to maintain the rooms at the desired temperature. This is accomplished with a maximum drop of about 10 F in the temperature of the ventilating current.

In summer the warm and humid air from out of doors is cooled and dehumidified to 64 F by contact with the cold water spray. In passing through the fan and duct work, the air increases in temperature from 2 to 7 F, depending upon the location of the room and upon the weather conditions. In circulating through the rooms, the air absorbs sufficient heat from the warm walls, glass, lights, and bodies of occupants to raise its temperature and at the same time to decrease the relative humidity to the desired degree. If the heat leakage into the rooms through the walls and glass is not sufficient to produce this effect, as is often the case in cool weather, the reheaters automatically furnish whatever additional heat is needed. Recirculation is used sparingly on account of the possibility of infection.

Success in applying these systems depends as much upon operation as upon design. The operation of the equipment, therefore, should not be entrusted solely to the hospital engineer. The system should be thoroughly checked by an expert twice every year and care should be taken to keep the apparatus clean, for if mud and fungus are allowed to accumulate in the air washer and Baudelot cooler tanks, musty and sour odors will pollute the air in the rooms.

COST DATA

The initial cost of the installation for the two nurseries alone was about $5,700.00. The addition of the third nursery brought the total cost to about $7,000.00 This sum represents the minimum cost of an installation of this type.

The itemized yearly cost for operating the system continuously at full capacity is as follows:

Electric power ($15.00 per year per kw of demand and $0.015 per kw-hr. of consumption)	$ 370.00
Steam ($0.10 per million Btu of demand and $0.60 per million Btu of heat)	670.00
Refrigeration ($1.25 per ton)	300.00
Water and compressed air	60.00
Supervision	300.00
Repairs and maintenance	150.00
Yearly operating cost (exclusive of depreciation and interest on capital investment)	$1,850.00

The unit costs shown are those charged to the Infants' Hospital by the Harvard Medical School Power Plant.

The operating cost is actually somewhat less than $1,600.00 because the special nursery for full term infants is not used continuously. The yearly operating cost for the two premature nurseries alone is about $1,200.00

SUMMARY

A four year study of the influence of air conditions on the growth and development of premature infants at the Infants' Hospital (Boston) showed that a mechanical air conditioning system of the central station type could be applied with marked success to the group care of these infants. The air conditioning requirements for the premature nurseries at the Infants' Hospital were found to be a ventilation rate of 25 air changes per hour, an air movement of 15 fpm over the beds, a relative humidity of 65 per cent, and environmental temperatures varying approximately from 100 to 76 F, depending on the general condition of the infant and its body weight. In order to meet these require-

ments, two conditioned nurseries were necessary, one about twice as large as the other. The special needs of congenitally diseased infants, particularly those of the 1.5 to 2.0 lb groups, were taken care of most satisfactorily by the use of open top electric incubators or heated beds placed inside the larger nursery.

The system should be designed for maximum indoor temperatures of 80 and 90 F in the larger and smaller nurseries respectively and for a dew-point of 70 F in winter and 60 F in summer. No recirculation allowance should be made in estimating the capacity of such systems. The problem of temperature control deserves especial attention, because premature infants are peculiarly sensitive to temperature changes.

The application of air conditioning to the premature nurseries of the Infants' Hospital was followed by a marked reduction in the incidence of diarrhea and in the mortality rate. Infections were reduced to a minimum and gains in body weight increased. These benefits are attributed to advances in medical treatment and improvement in nursing and general care, as well as to the installation of the air conditioning system described.

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DISCUSSION

F. C. HOUGHTEN: As engineers we should welcome this paper as marking the beginning of a new epoch in the progress and application of air conditioning as an aid to human comfort and well being.

Professor Yaglou, a former member of our Research Laboratory staff and co-worker in the development of the comfort chart, and his associates at Harvard, have applied air conditioning in the care of premature infants with phenomenal success. In fact, they have succeeded in reducing what has been generally accepted as the normal death rate among such patients by something over 50 per cent. That is certainly a phenomenal success and one which warrants the highest appreciation of civilized society. As engineers, we should remember that it was proper knowledge of air conditioning which made this possible.

It seems to me almost axiomatic that air conditioning should be very effective in the treatment of many diseases. Any disease that results in a fever condition, represents a condition of either an excessive rate of heat production in the body or failure in heat elimination from the body. The results of the

Laboratory have shown that atmospheric conditions—that is temperature, humidity and motion of the air and other factors—have a very marked effect on the rate of heat dissipation and the way in which that heat is eliminated; that is whether it is dissipated as sensible heat or by the evaporation of perspiration. It would appear to be a very short-sighted policy to fail to take into consideration the atmospheric environment of a person so afflicted. It would seem entirely logical that the atmospheric environment should be of much greater importance than the food and medicine which he takes into his digestive system.

A great many men in the medical profession are now thinking of the possibility of applying atmospheric conditioning in relation to treatment of illness and a single incidence of this trend came to our attention at the Laboratory recently. Certain Pittsburgh doctors acquainted with the work of our Laboratory along the line of air conditioning and its relation to human well-being conceived the possibility of treating a certain serious illness which has heretofore baffled the skill of the medical profession by air conditioning. The Laboratory made available the air conditioned rooms on certain days when they were not otherwise in use. A few patients were treated with what seems, to date, a considerable degree of success. It seems undesirable by these doctors to announce the nature of the disease or the treatment until they are more certain of the results but apparently complete cure or marked improvement has resulted in the cases treated.

The benefits derived from proper air conditioning for premature infants and the case just cited may be taken as an index of many other possibilities and as air conditioning engineers, we should bring these possibilities to the attention of the medical profession.

T. J. Duffield: I think the whole crux of this paper is contained in the last paragraph. Here we have four valid criteria by which one might very properly judge improvement in the health of these very sensitive reactors to air conditions and in all four of them definite gains are demonstrated. As the authors say elsewhere, it is inescapable to conclude that the air conditioning has had a very important place in bringing about the results that are reported.

The authors point out that the comfort chart that was designed at the Research Laboratory of the Society is not applicable to the premature infant, and that is only logical, of course. I noticed, too, that these authors are very conservative and modest in their claims for the effects of air conditioning and do not attempt to ascribe the entire benefits that these children received to the improvement of air conditions, but they do point out that an improvement has been made along the lines of medical treatment in the nursing and general care.

This paper without doubt marks a new epoch in our application of air conditioning for the improvement of health. With the exception of the information from Holland, where allergic diseases have been successfully treated and prevented by air conditioning, this is the first bit of scientific evidence we have had which directly ties up health, other than the passing physiological conditions, to improvement in air conditions.

Physicians in several parts of the country are seriously considering the possible application of air conditioning in the treatment of several serious diseases and it is a splendid indication that there has been awakened among the medical fraternity in the United States, what we might call "air-mindedness," in this case applying to air conditioning rather than to aviation.

I consider that this is an epoch-marking paper and that in the near future we shall see a much wider application of air conditioning to the treatment of diseases than we have had in the past.

HADAR GILLE (SWEDEN): We have used your charts for air conditioning in Sweden. A doctor wished to make some blood tests and he at first tested it on himself. He especially wished to test the difference in the contents in the blood of organisms with and without fever. These tests were made on a horse as a horse has the same temperature control as a human being. The control was obtained by means of sweating. The doctor asked me if air temperature had any influence on the body temperature and I referred him to the data obtained at the A.S.H.V.E. Laboratory on the effect of extreme temperatures on the human body. It was astonishing that we could determine the number of hours required for the horse's body to attain the desired temperature. We could accurately determine the necessary air condition to get the result, a temperature of 42 C was obtained by means of the figures you have determined for the human body.

I received a letter stating that the condition in the blood of the horse was as if it had got the fever from a disease. As you know fever is a weapon against disease. Why should it not be possible to use air conditioning for this purpose? I think it could be made a good weapon in the fight against disease.

E. C. EVANS: I would like to see research carried on somewhere, preferably at our own Laboratory to better the comfort and health conditions of those who because of their tubercular trouble are forced to reside in the hot and dry climate of our desert lands, where they are able to take advantage of the curative powers of the natural atmosphere which in some way that is mysterious to us does prolong the life of those affected with this sickness. Many of these persons are well able to afford air conditioning in their homes and some of them want cool air.

It is an easy matter to provide cool air, but what will be the effect on health. In other words, what effect does the rising humidity have on the production of mucous which condition is practically absent in the dry air under natural conditions. There is room for much research on this subject. Certainly we should know how far we can reduce the dry-bulb temperature when we would naturally have an increase of humidity. Therefore we must know the values to be expected as they relate to health.

A. S. LANGSDORF: A recent number of *Science* carried an account of the work of two physicians, who have recently carried on extensive experiments designed to produce by electrical treatment local artificial fevers in different parts of the human body.

In the School of Medicine of Washington University, some of the members of the faculty are working on artificially induced fevers, using a high-frequency electrical discharge. When I was asked about some of the engineering features of the high frequency apparatus, I raised the question whether the observed temperature rise to 105 F might not be in error, because mercury thermometers were used to measure the internal temperature. In all such cases an alcohol thermometer should be used for accuracy.

No. 868

AIR INFILTRATION THROUGH VARIOUS TYPES OF WOOD FRAME CONSTRUCTION

By G. L. Larson[1], D. W. Nelson[2] and C. Braatz[3], Madison, Wis.

MEMBERS

The results of cooperative research between the University of Wisconsin and the American Society of Heating and Ventilating Engineers

Introduction

FOR the past three years, a program of cooperative research has been under way sponsored by the *University of Wisconsin* and the American Society of Heating and Ventilating Engineers for the purpose of determining the relative air tightness of various types of building construction. The initial work of this program was confined to brick wall construction, the results of which are presented in the paper entitled, Air Infiltration Through Various Types of Brick Wall Construction, p. 99.

In the Summer of 1929, work was begun on a series of tests pertaining primarily to wood frame construction under a cooperative agreement between the Society, the University and the *National Lumber Manufacturers' Association.* It is the results of the latter program with which this paper is concerned.

Description of Test Apparatus

The test apparatus shown in Fig. 1 consists briefly of the pressure chamber *A,* and the collecting chamber *B,* between which the panel to be tested is secured by means of *C*-clamps. Air tight seals are obtained between the two sides of the panel frame and chambers *A* and *B* by means of a sponge rubber gasket attached to the perimeter of the chamber openings. Artificial wind pressure is produced by a small motor-driven blower, shown at the extreme left of Fig. 1. The blower is in communication with the pressure chamber through an adjustable damper *E* by means of which the pressure drop through the wall is controlled. Other control dampers are provided at *D* and on the intake to the blower itself. The pressure difference in chambers *A* and *B,* which is the pressure drop through the wall panel, is measured with an inclined draft gage, *F.*

The amount of air passing through a wall panel is measured by a set of interchangeable orifices ranging in size from 3/8 in. to 6 in. in diameter, mounted on the end of orifice box *C.* The pressure drop through the orifice is determined

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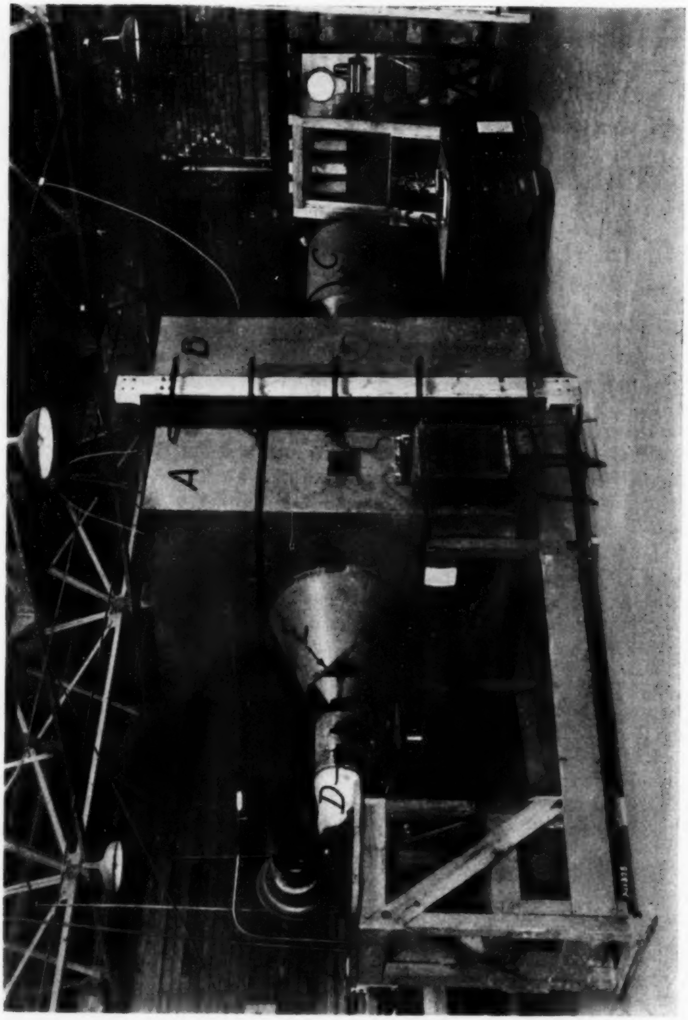

FIG. 1. GENERAL LAY-OUT OF TEST EQUIPMENT

by a Wahlen gage, *G*. To facilitate examination of a wall panel after it has been clamped in place, without disturbing the seal between the test machine chambers and the wall frame, a manhole is provided in each chamber on the side opposite that shown in Fig. 1.

DESCRIPTION OF TEST PANEL FRAMES

The test panels, described briefly in Tables 1 and 2 and in detail in the appendix of this paper, were built into rectangular frames constructed of four 4 × 8 air-dried Douglas fir timbers, in the manner shown in Fig. 15. The frame members were secured at the four corners by means of 6 × 6 × ½ in. angle irons, and the joints at these corners were thoroughly sealed with a plastic calking compound.

All construction work was done by a local contractor, under the supervision of L. V. Teesdale of the *U. S. Forest Products Laboratory,* who also passed upon the suitability of all construction material, with reference to specifications drawn by the *National Lumber Manufacturers' Association.* Every effort was made to have all construction comparable to that in actual building practice.

TEST PROCEDURE

Each test panel was subjected to wind pressure ranging from about 5 to 30 mph, for each of the test conditions as outlined in Tables 1 and 2. Except in the case of Panels *2B* and *3B,* the crack between the frame members and the test panel perimeter was completely sealed with plastic calking compound, leaving only the surface of the panel, 51 sq ft, open to leakage.

With the exception of the panel described in Item 34 of the appendix, all seasoning periods took place indoors at a temperature of approximately 70 F. The seasoning period for panel *6D,* Item 34, consisted for the most part of sub-zero weather together with one light snowfall and several days of sunshine.

DESCRIPTION OF TABLES 1 AND 2

A brief description of the different constructions built and tested is given in Tables 1 and 2. Cross-sections of the constructions listed in Table 1 are shown in Fig. 2 and for those listed in Table 2 in Fig. 3. More complete descriptions of the constructions built are given in the appendix. The description in the appendix for any particular panel number may be located by referring to the number in the column headed *Index to Specifications.*

Tables 1 and 2, in addition to giving a brief description of the constructions made and tested, list the standing periods from the date of construction to the date of test. Figs. 4 to 15 are photographs of typical wall constructions that were built. The columns headed *Index to Figures* in Tables 1 and 2 list the panel constructions for which photographs are included in this paper. These are the numbers appearing before the dash.

DISCUSSION OF RESULTS

The results of the tests on each one of the constructions made are given in Tables 1 and 2 under the heading of *Air Infiltration in Cubic Feet per Hour per Square Foot of Wall.* The results are given at wind velocities from 5 to 30 mph by 5-mile intervals. These wind velocities correspond to drops in pressure in inches of water across the wall as observed during the tests. The relation

TABLE 1. DESCRIPTION OF TEST PANELS AND SUMMARY OF RESULTS OF AIR INFILTRATION TESTS

Panel No.	Description of Construction: Outside Construction: Sheathing	Bldg. Paper	Siding	Inside Construction	Standing Period in Days	Air Infiltration in Cubic Feet Per Hour Per Square Foot of Wall: Wind Velocity—Miles Per Hour: 5	10	15	20	25	30	Panel No.	Index: To Figures	To Spec.
1A	1x6 Green Dressed and Side Matched No. 1 Common	None	None	None	29	16.3	47.0	81.2	115.0	149.9	188.0	1A	4	3
1A1	As in 1A except knot hole plugged	None	None	None	29	16.3	46.5	78.6	111.8	145.8	182.0	1A1	16	3
1B	1x6 Air Dried Butt-Edged No. 1 Common	Bldg. Paper B	1x10 Bevel Siding Air Dried	None	70	0.04	0.13	0.28	0.45	0.62	0.78	1B	20	22
1B1	As in 1B	As in 1B	Paint Added to Siding in 1B	None	6	0.03	0.10	0.18	0.27	0.35	0.43	1B1	20, 22	22
1C	As in 1B	As in 1B	As in 1B1	Wood Lath and 3 Coats Gyp. Plaster	68	0.03	0.09	0.16	0.23	0.29	0.33	1C	22, 24	23
1D	1x6 Air Dried Dressed and Side Matched No. 1 Common	None	¾x2 Expanded Metal 1⅛ in.—3 Coats Stucco	None	29	0.02	0.05	0.08	0.12	0.14	0.15	1D	25	15
2A	1x6 Air Dried Dressed and Side Matched No. 1 Common	None	None	None	28	2.4	8.0	15.3	23.3	32.5	42.3	2A	5,–16, 25	4
2B	As in 2A	Bldg. Paper A	1x6 Drop Siding Air Dried	None	14	0.04	0.13	0.28	0.45	0.62	0.78	2B	7,–20, 21	6
2B1	As in 2A	As in 2B	Paint Added to Siding in 2B	None	6	0.03	0.12	0.27	0.43	0.59	0.75	2B1	20	6
3A	1x6 Air Dried Dressed and Side Matched No. 1 Common	None	None	None	28	1.3	4.6	9.3	15.1	21.5	28.8	3A	6,–16, 25	4
3B	As in 3A	Bldg. Paper A	1x6 Bevel Siding Air Dried	None	14	0.04	0.13	0.28	0.45	0.62	0.78	3B	7,–20, 21	6
3B1	As in 3A	As in 3B	Paint Added to Siding in 3B	None	6	0.02	0.06	0.11	0.17	0.22	0.27	3B1	20	6
3C	1x4 Boards on 5-in Centers No. 2 Common	None	5/2-16 in. Red Cedar Shingles—5-in. Exposed	None	69	13.4	39.5	69.5	101.1	135.0	172.0	3C	19, 25	16
3D	As in 3C	Bldg. Paper B	As in 3C	None	69	0.07	0.22	0.38	0.54	0.70	0.84	3D	19	21
4A	1x6 Air Dried Dressed End and Side Matched No. 1 Common	Bldg. Paper A Studs Vert.	None	None	29	0.05	0.16	0.31	0.50	0.69	0.90	4A	8,–18	5
4A1	As in 4A	Cut Out	None	None	None	20.0	48.5	80.3	114.1	148.2	184.5	4A1	16	5
4B	1x8 Air Dried Ship-lap No. 2 Common	None	5/2-16 in. Red Cedar Shingles — 5-in Exposed	None	14	2.1	7.5	15.3	24.8	34.9	45.3	4B	9, 10–19	17
4D	None	Bldg. Paper B Studs Vertical	1x6 Air Dried Select Drop Siding Painted	None	4	0.06	0.15	0.23	0.31	0.38	0.45	4D	....	31
4E	None	As in 4D	As in 4D	1x6 Sheathing End and Side Matched	6	0.06	0.15	0.23	0.31	0.38	0.45	4E	24	32
4F	1x6 Air Dried Dressed End and Side Matched No. 1 Common	None	None	None	None	3.6	13.4	28.3	45.5	63.8	82.7	4F	16, 25	33

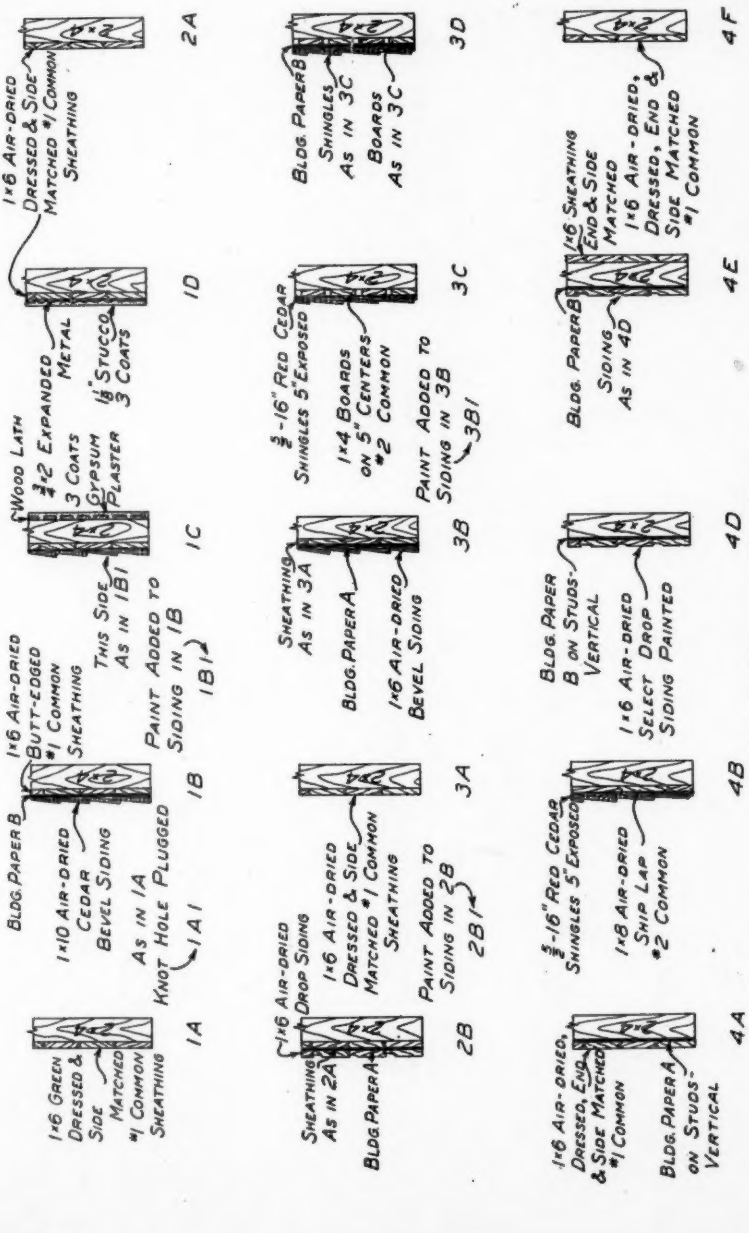

FIG. 2. CROSS-SECTIONS OF PANELS

TABLE 2—DESCRIPTION OF TEST PANELS AND SUMMARY OF RESULTS OF AIR INFILTRATION TESTS

Panel No.	Description of Construction				Standing Period In Days	Air Infiltration in Cubic Feet Per Hour Per Square Foot of Wall							Index	
	Outside Construction			Inside Construction		Wind Velocity—Miles Per Hour						Panel No.	To Figures	To Spec.
	Sheathing	Bldg. Paper	Siding			5	10	15	20	25	30			
6A	Insulation A Sheathing Board	None	None	None	30	3.8	9.8	16.1	22.5	29.2	36.4	6A	11,–16	7
6B	None	None	Corrugated Steel	None	14	9.0	26.2	45.4	65.1	85.2	106.7	6B	12,–25	9
6C	1x8 Air Dried Ship-lap No. 2 Common	None	24 in. Red Cedar Shingles 11 in. Exposed	None	14	7.2	23.1	43.8	67.1	90.9	116.7	6C	19	19
6D	None	None	1x10 Boards — Spaced 1x4 Battens — Paint	None	14	16.3	49.0	86.9	126.5	165.8	206.0	6D	13,–25	34
7A	Insulation B Sheathing Board	None	None	None	30	1.6	5.0	9.1	13.2	17.8	22.8	7A	16	8
7B	1x6 Boards on 11 in. Centers No. 2 Common	None	24 in. Red Cedar Shingles 11 in. Exposed	None	15	22.1	68.0	122.5	180.4	240.6	307.1	7B	9, 10–19, 25	18
7C	As in 7B	Bldg. Paper B	As in 7B	None	67	0.02	0.07	0.13	0.19	0.23	0.25	7C	18	20
7D	1x6 Air Dried Ship-lap	Bldg. Paper B	5/2-16 in. Red Cedar Shingles — 7½ in. Exposed	None	7	0.03	0.10	0.17	0.25	0.33	0.42	7D	19, 22	29
7E	As in 7D	As in 7D	As in 7D	Wood Lath and 3 Coats Gyp. Plaster	14	0.02	0.06	0.12	0.18	0.25	0.31	7E	22, 24	30
9A	None	None	None	Wood Lath and 3 Coats Gyp. Plaster	15	0.02	0.08	0.17	0.28	0.40	0.52	9A	14–23	10
9B	None	None	None	As in 9A Plus Decorative Paper	10	0.01	0.05	0.10	0.15	0.20	0.26	9B	23	11
9D	None	None	Insulation A Lath and Stucco	None	29	0.07	0.27	0.39	0.56	0.73	0.90	9D	25	14
9E	1x6 Air Dried End and Side Matched No. 1 Common	None	None	None	2	3.6	13.4	28.3	46.3	64.7	83.8	9E	16, 17, 25	24
9F	As in 9E	Poor Grade Resin	None	None	None	0.4	1.5	2.9	4.3	5.3	5.9	9F	17, 18	25
9G	As in 9E	As in 9F	1x6 Drop Siding Painted	None	13	0.01	0.05	0.10	0.14	0.18	0.21	9G	17	26
10A	None	None	None	Metal Lath and 3 Coats Gyp. Plaster	14	0.03	0.12	0.23	0.35	0.47	0.60	10A	14,–23	12
10B	None	None	None	As in 10A Plus Sizing and Flat Paint	10	0	0	0	0	0	0	10B	23	13
10C	None	None	None	1x6 Sheathing End and Side Matched	7	8.3	26.5	48.8	72.0	96.7	122.8	10C	16, 25	27
10D	None	None	Corrugated Steel	As in 10C	4	1.2	4.0	8.1	12.7	17.4	22.2	10D	24	28
10E	None	None	As in 10D	None	63	1.4	4.7	9.1	13.8	18.5	23.6	10E	25	28

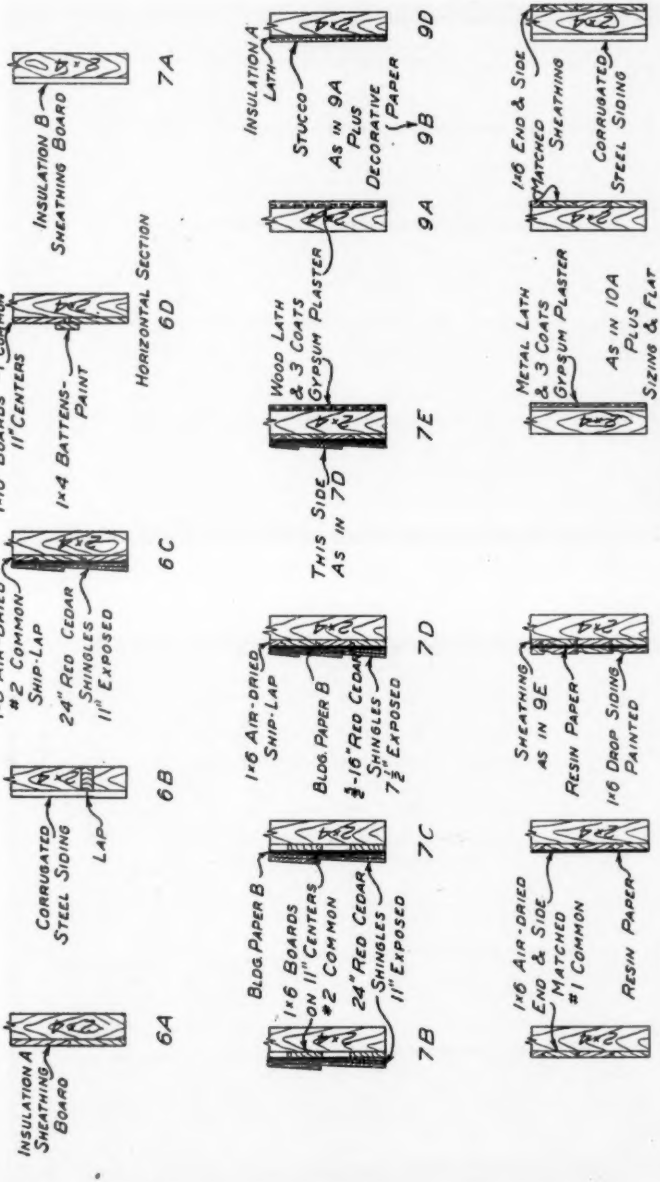

FIG. 3. CROSS-SECTIONS OF PANELS

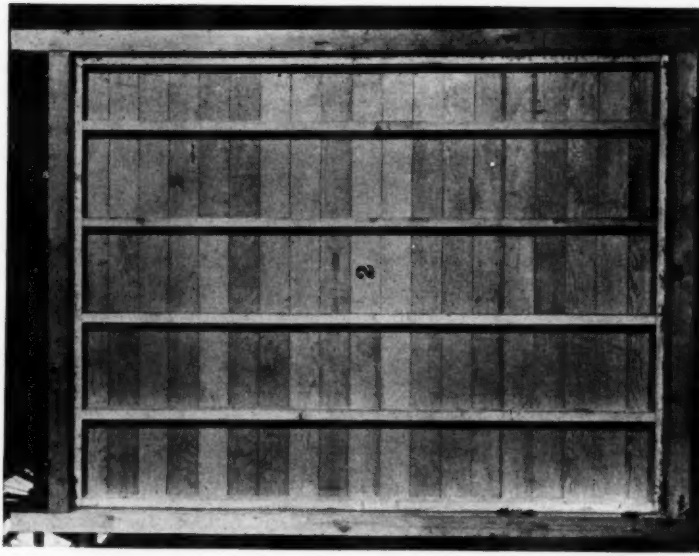

FIG. 5. INTERIOR OF PANEL NO. 2*A*—AIR DRIED SHEATHING

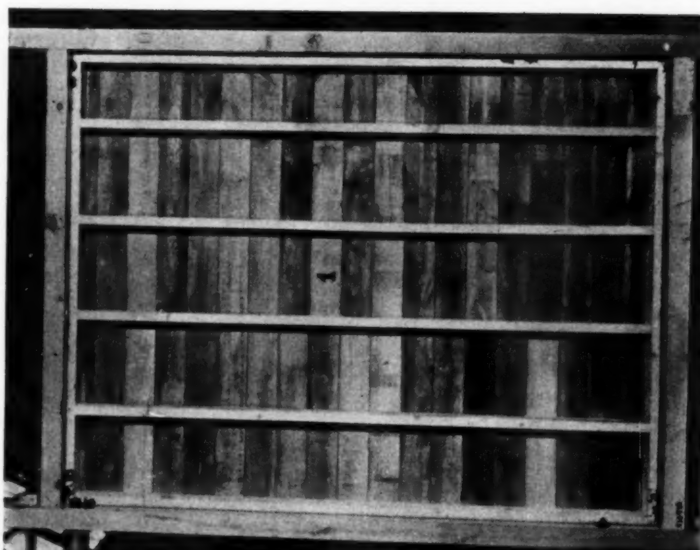

FIG. 4. INTERIOR OF PANEL NO. 1*A*1—GREEN SHEATHING

between the drop in pressure in inches of water and the wind velocity is as follows:

Drop in Pressure in Inches of Water......	0.012	0.048	0.108	0.192	0.300	0.431
Wind Velocity in Miles per Hour	5	10	15	20	25	30

The test results are shown graphically in Figs. 16 to 25 grouped to show comparisons between various important features. The column headed ***Index to Figs.*** in Tables 1 and 2 shows the figures in which the test results of each panel number appear. The figure number before the dash is that of the photograph of the wall construction, if any is included, and the numbers after the dash refer to the figures showing the test results graphically. Not all of the test results have been plotted but only those most useful in making comparisons. Tables 1 and 2, however, show the results of all tests and any additional comparisons may be made by referring to the results as tabulated there.

INFILTRATION THROUGH SHEATHING ONLY

The results of tests on the various sheathings are shown grouped together in Fig. 16. The four tests on air-dried end and side matched sheathing show considerable variation. The average of the four tests shows a leakage of 46.4 cfh per square foot of wall at 15 mph. Of these four end and side matched sheathing panels, 4*F*, 9*E* and 10*C*, were of a uniform material and the average of these would be 35.1 cfh per square foot at 15 mph. As compared to this average, the leakage through Panel 4*A*1 which was of a less uniform material was 80.3 cfh. The lowest leakage secured for end and side matched sheathing was 28.3 cfh per square foot, the value obtained in tests of Panels 4*F* and 9*E*.

The air-dried side matched sheathing on Panel 2*A* showed a leakage of 15.3 cfh per square foot and on Panel 3*A* a leakage of 9.3 cfh per square foot of wall both at 15 mph. This is an average leakage at this wind velocity of 12.3 cfh per square foot. This figure of 12.3 cu ft for side matched sheathing is to be compared with the figure of 35.1 cu ft for end and side matched sheathing for the same material. In the case of side matched sheathing, all end joints are butted on the studding, whereas many joints with end and side matched sheathing come in between studding. The increase in leakage of end and side matched sheathing over side matched sheathing is considered to be due to greater leakage through the end joints. Figs. 5 and 6 show the air dried side matched sheathing used in Panels 2*A* and 3*A*. The end and side matched sheathing of Panels 4*F*, 9*E* and 10*C* was of this same material.

The leakage through green side matched sheathing, Panel 1*A*1, was 78.6 cfh per square foot at 15 mph. This is slightly less than the leakage through air dried end and side matched sheathing of like material. The comparatively low leakage secured indicates that even with green lumber the shrinkage is not ordinarily enough to pull the tongues and grooves apart and allow the free passage of air. A photograph of Panel 1*A*1 is shown in Fig. 4.

The leakage through fibre board as found in tests of Panels 6*A* and 7*A* was 12.6 cfh per square foot at 15 mph. This is about the same as through air dried side matched sheathing as determined on Panels 2*A* and 3*A*. The variation in the infiltration found in the two tests of fibre board was due to a variation in the width of the horizontal joints. The sheets were closely fitted together on Panel 7*A*, but a section of the joint on Panel 6*A* had a crack of an

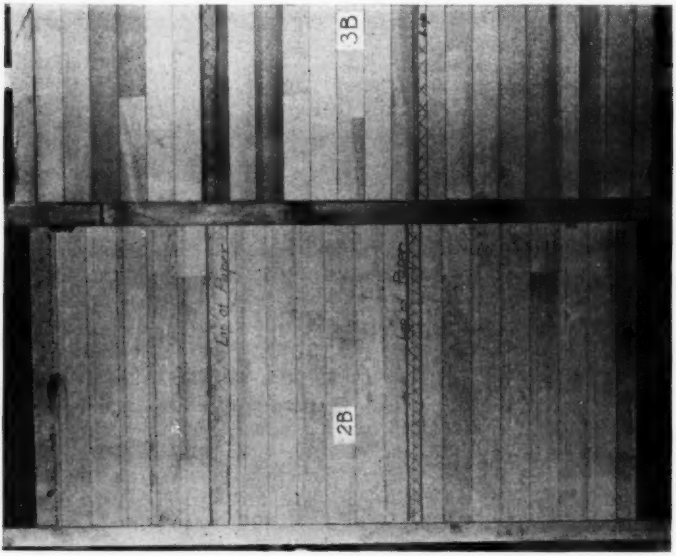

FIG. 7. EXTERIOR OF PANEL NO. 2*B*, SHEATHING, PAPER AND DROP SIDING. EXTERIOR OF PANEL NO. 3*B*, SHEATHING, PAPER AND BEVEL SIDING

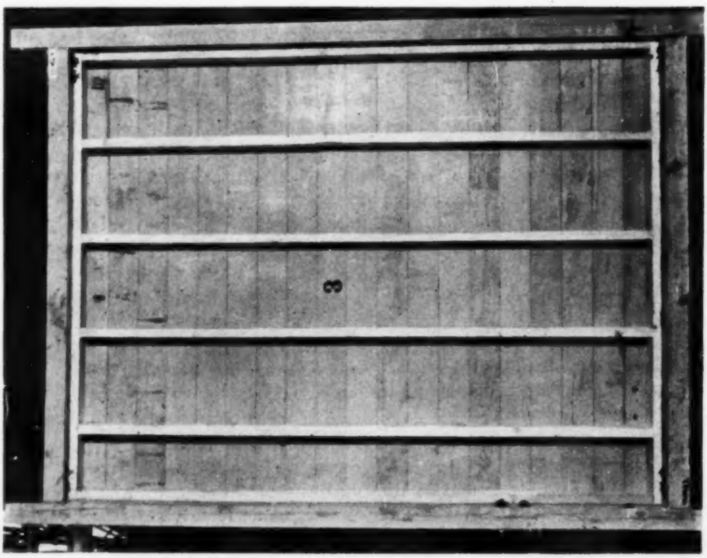

FIG. 6. INTERIOR OF PANEL NO. 3*A*, AIR DRIED SHEATHING

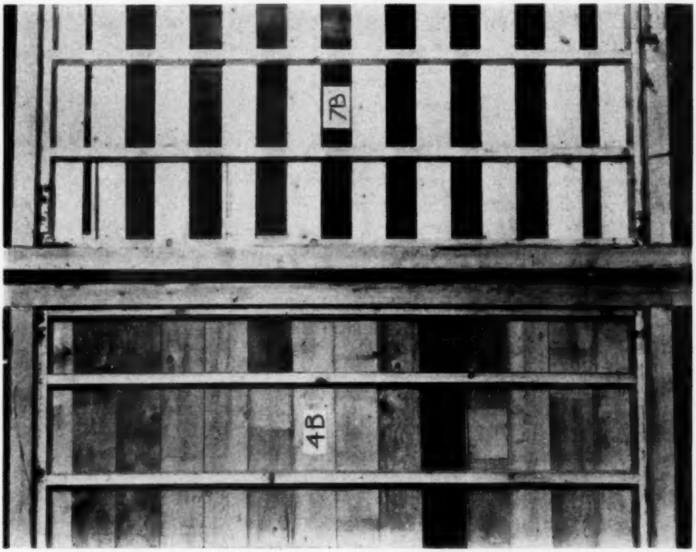

FIG. 9. INTERIOR OF PANEL NO. 4*B*, 16-IN. SHINGLES ON SHIPLAP. INTERIOR OF PANEL NO. 7*B*, 24-IN. SHINGLES ON SPACED BOARDS

FIG. 8. INTERIOR OF PANEL NO. 4*A*—AIR DRIED SHEATHING

appreciable width. Fig. 11 shows a photograph of Panel *6A*. When the horizontal joints were sealed the leakage for Panels *6A* and *7A* was the same. The leakage through a wall of fibre board depends largely on the fit of the horizontal joints and may be considerable with careless workmanship.

ADDITION OF BUILDING PAPER, DROP SIDING AND PAINT TO SHEATHING

Fig. 17 shows the infiltration resulting from tests of the various steps in the building of the outside construction of a wall. The leakage through the sheathing only in Panel *9E*, which was end and side matched was 28.3 cfh per square foot of wall at 15 mph. The application of ordinary resin-sized building paper over the sheathing with the seams nailed at 7-in. intervals reduced the leakage in Panel *9F* to 2.9 cfh per square foot of wall at 15 mph. The addition of drop siding and paint in Panel *9G* reduced the leakage to a negligibly small amount. This reduction is due not only to the resistance to air flow of the painted drop siding but also to the increased effectiveness of the sheathing paper because of being clamped between two thicknesses of boards.

VARIOUS APPLICATIONS OF SHEATHING PAPER

That the effectiveness of sheathing paper depends to a considerable extent on the method of application is shown by the curves in Fig. 18. Paper nailed on the sheathing with the laps nailed at approximately 7 in. intervals in Panel *9F* showed a leakage of 2.9 cfh per square foot of wall at 15 mph. Paper applied vertically between the studs and sheathing in Panel *4A* showed a leakage of 0.31 cfh per square foot at 15 mph. In Panel *7C*, paper was placed between two thicknesses of spaced boards and shingles which by themselves showed an extremely high leakage. The extremely low leakage of 0.13 cfh per square foot at 15 mph found in the test of Panel *7C* is attributable to the effectiveness of the paper when tightly clamped between two thicknesses.

The paper used on Panel *9F* was a poor grade of resin paper. In Panel *9G* drop siding and paint were added which reduced the leakage to a negligibly small amount just as was the case with a good grade of building paper. This is attributable to the reduction in leakage at the laps due to the clamping of the paper between the two thicknesses of sheathing and drop-siding. In the average building construction, a good grade of sheathing paper would probably have a greater resistance to air flow than would a poor paper because of less tearing in application due to its greater tensile strength. Also it probably would maintain its resistance to air leakage better over a period of years against the effects of aging and weathering.

In the construction of farm and other shelter buildings the application of paper in the proper way would seem to be worth while. There are two ways of applying the paper between the studding and the sheathing, horizontally and vertically. Horizontal application between studding and sheathing would correspond in leakage to the results secured in test of Panel *9F* where the sheathing paper was applied horizontally on the outside of the sheathing and nailed at intervals. Vertical application between studding and sheathing was made in Panel *4A* and showed considerably less leakage than horizontal application.

FIG. 10. EXTERIOR OF PANEL NO. 4*B*, 16-IN. SHINGLES ON SHIPLAP. EXTERIOR OF PANEL NO. 7*B*, 24-IN. SHINGLES ON SPACED BOARDS

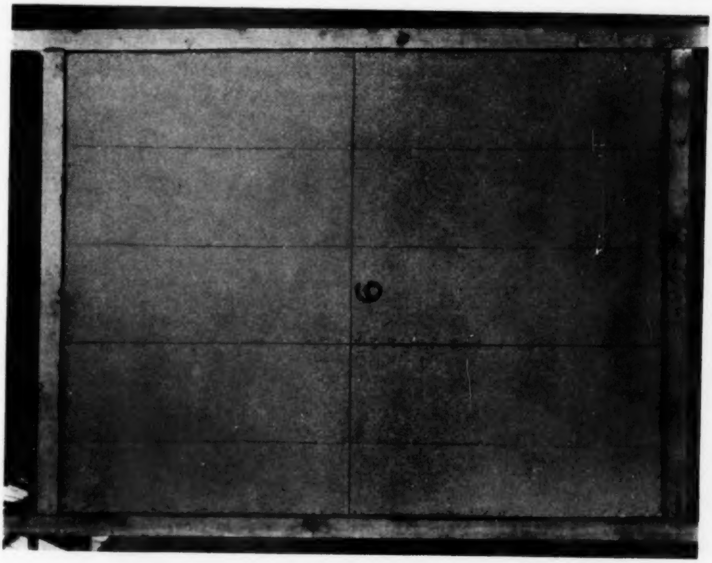

FIG. 11. EXTERIOR OF PANEL NO. 6*A*—FIBRE SHEATHING BOARD

FIG. 12. EXTERIOR OF PANEL NO. 6B—CORRUGATED STEEL SIDING

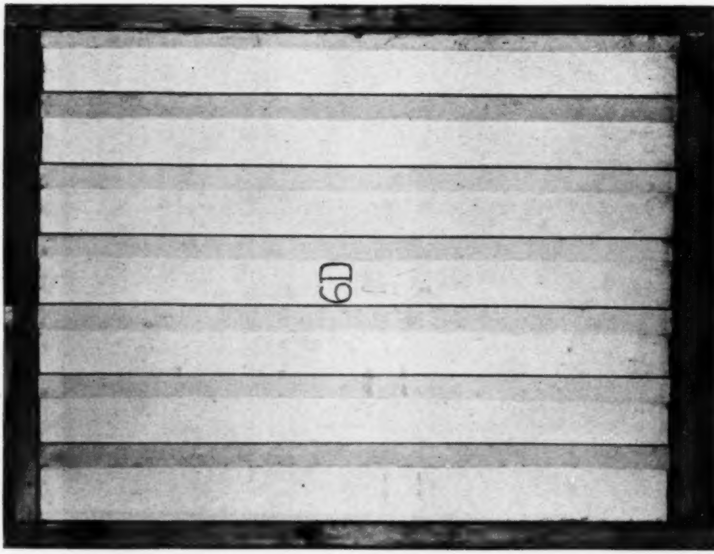

FIG. 13. EXTERIOR OF PANEL NO. 6D—1x10 SPACED BOARDS WITH BATTENS

AIR INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

Fig. 19 shows the results of tests on various shingle constructions. The greatest leakage was secured with 24-in. shingles nailed to 1x6 boards spaced on 11-in. centers in Panel *7B*. This leakage was 122.5 cfh per square foot of wall at 15 mph. The corresponding construction for 16-in. shingles and 1x4 boards on 5-in. centers in Panel *3C* showed a leakage of 69.5 cfh per square

FIG. 14. EXTERIOR OF PANEL NO. 9*A*, PLASTER ON WOOD LATH. EXTERIOR OF PANEL NO. 10*A*, PLASTER ON METAL LATH

foot. In both of these constructions, the spaced boards act merely as pieces to nail the shingles to and play practically no part in the stopping of air leakage.

The replacing of the spaced boards with shiplap in the case of 24-in. shingles in Panel *6C* reduced the leakage to 43.8 cfh per square foot at 15 mph. The corresponding value for 16-in. shingle construction in Panel *4B* was 15.3 cfh per square foot. The use of shiplap in shingle roof and side wall construction would seem to be advisable in the reduction in infiltration.

The application of sheathing paper to a 16-in. shingle construction using

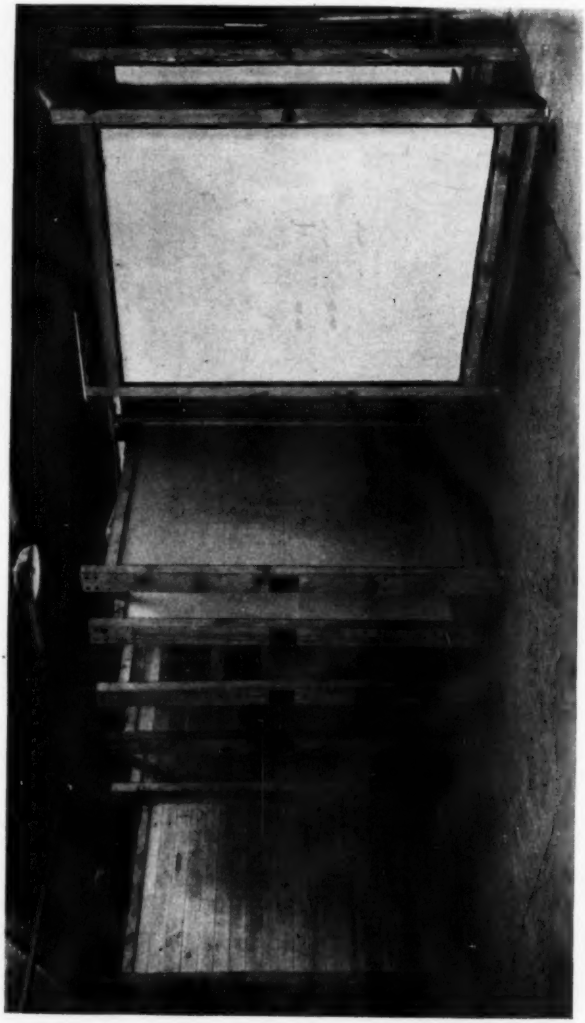

FIG. 15. ASSEMBLY OF TEST PANELS

spaced boards in Panel *3D* and to a 16-in. shingle construction using shiplap in Panel *7D* reduced the air infiltration to the negligibly small amount of less than one-half cubic foot per hour per square foot of wall. The great reduction secured by the use of paper in a shingle wall would make its use advisable in any construction where infiltration is objectionable. One construction where the use of paper would be highly desirable is in shingle roof construction on residences.

ADDITION OF PAINT TO SIDING

The infiltration of air through a wall having building paper properly applied is negligibly small. This amount was 0.28 cfh per square foot of wall at 15 mph for Panels *1B, 2B* and *3B*. Fig. 20 shows the results of these tests plotted to a very much enlarged scale. The application of paint to the siding on these panels resulted in a further reduction.

In the case of Panel *2B*1, having drop siding, the reduction was very slight, but was an appreciable amount in Panels 1*B*1 and 3*B*1 having bevel siding. This indicates that paint seals the joints in bevel siding better than in drop siding. This reduction in either case is extremely small since the sheathing paper clamped between the sheathing and siding by itself is very effective in preventing air infiltration. This reduction in leakage due to the addition of paint probably is not permanent since it would take very little movement due to weathering or to changes in humidity to break the paint film sufficiently to allow air to reach the sheathing paper quite freely as compared to its passage through the paper layer. Paint applied to a weathered siding would of course reduce the size of the cracks at joints in the siding.

A STUDY OF GOOD AND POOR CORNER CONSTRUCTIONS

In the case of Panels *2B* and *3B* tests were first made with the joint between the wall and the frame calked on only three sides. On the fourth side, the joint at the depth of the sheathing only was calked. This allowed air to get under the siding and under the paper from this uncalked end joint. This is considered to be approximately the condition at the corner of a frame building or against window or door openings with poor construction. Here there is a chance for air to get under the trim, then to get under the siding and seek out openings in the layer of paper. Unless the paper is very carefully applied, there would also be the chance for air to find its way under it directly at the corner or at the window or door frame.

Fig. 21 shows the results of these tests with one edge of the siding and paper not sealed. The results on Panel *3B* showed a leakage of 1.70 cfh per square foot of wall at 15 mph for a wall equipped with bevel siding. The corresponding results on Panel *2B* equipped with drop siding was 0.45 cu ft. This indicates that drop siding, since it presses down over the entire paper surface quite effectively, prevents air from getting under the paper from an end joint such as occurs at the corner of a building or at window or door frames. The bevel siding, since it makes only line contact with the paper, is not so effective in preventing air from getting under the paper. This points to the need for carefully wrapping the paper around the corner of a building and for the making of a tight seal in the paper layer at window and door openings. In the case of bevel siding with the best paper application at such locations, there is still the chance for air to travel horizontally in the air spaces under

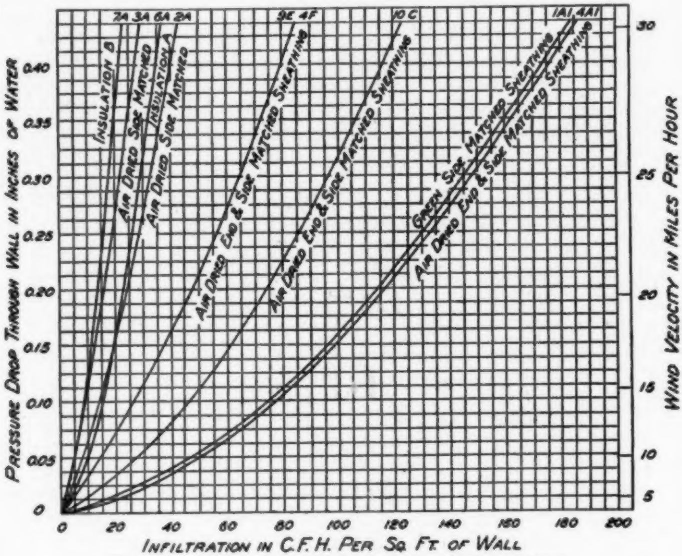

FIG. 16. INFILTRATION THROUGH SHEATHING ONLY

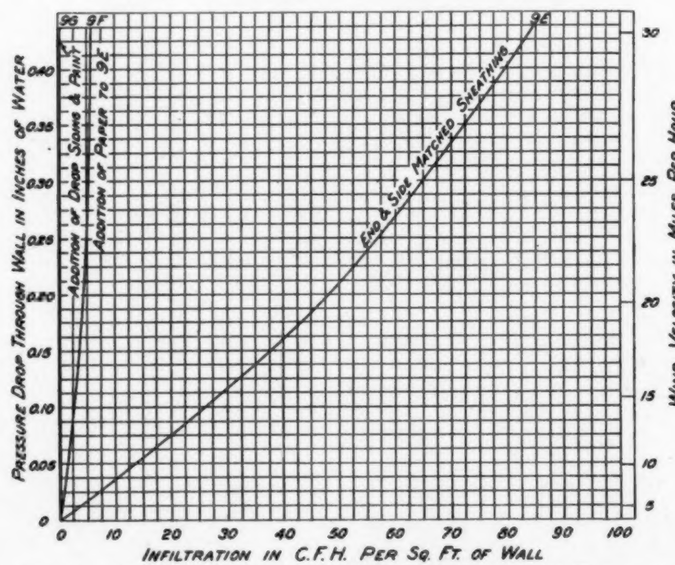

FIG. 17. ADDITION OF BUILDING PAPER, DROP SIDING AND PAINT TO SHEATHING

the siding and seek out defects in the paper. This calls for careful application not only at corners at openings but over the entire wall surface. It is also important that laps in the paper layer come under lines of contact when bevel siding is to be applied.

When the fourth side of Panels *2B* and *3B* was sealed, the leakage was reduced to the negligibly small amount of 0.28 cfh per square foot. Fig. 7 shows a photograph of these two panels.

Addition of Plaster to Walls Having Sheathing Paper

The results of tests on Panel *1B1* made of sheathing, paper and bevel siding and Panel *7D* consisting of shiplap, paper and shingles are plotted in Fig. 22 to a greatly exaggerated scale. The leakages for these two constructions were 0.18 and 0.17 cfh per square foot at 15 mph. These are extremely small leakages. The addition of plaster to these two constructions in Panels *1C* and *7E* resulted in a further reduction of this negligibly low leakage.

Tests on plaster constructions alone in Panels *9A* and *10A* showed the negligible leakage of 0.17 and 0.23 cfh per square foot at 15 mph.

Having practically air tight layers such as building paper and plaster at two points in a wall should reduce air movements within the wall and thereby exert a beneficial effect on heat losses by transmission.

Addition of Wall Paper and Paint to Plaster

Plaster by itself allows only a negligible amount of air leakage. The results of tests of plastered Panels *9A* and *10A* are plotted to an exaggerated scale in Fig. 23. The infiltration averaged 0.20 cfh per square foot at 15 mph. The tests were made with no cracks in the plaster and the panels were sealed completely on all four sides. Had cracks been present or had imperfect sealing been made at an edge of the plaster sheet corresponding to poor sealing at the baseboard or at a window opening in actual construction, the leakage would have been much greater. To obtain the effectiveness of plaster against air leakage good workmanship includes the proper sealing of the plaster sheet at the baseboard and window trim. The addition of wall paper in Panel *9B* to the plaster in Panel *9A* resulted in a considerable although unimportant reduction in air leakage. The addition of sizing and flat wall paint in Panel *10B* to the plaster in Panel *10A* made the wall practically air tight. The leakages of the plain plaster walls were negligibly small. The application of paint or paper under such circumstances would have to be justified by other reasons than a reduction in air infiltration. Probably the application of paint or paper to a badly cracked plaster wall would have a favorable effect on air leakage.

Infiltration Through Various Types of Compound Walls

A compound wall for the purposes of this paper is defined as one that has a finished construction on both sides of the studding. The results of tests on the four compound walls that were built are shown in Fig. 24. The three panels *1C*, *4E* and *7E* all had in their construction sheathing paper and plaster, although different types of siding were used. The leakages secured were 0.16, 0.23 and 0.12 cfh per square foot at 15 mph or an average for the three walls of 0.17 cu ft.

Panel *10D* contained neither sheathing paper nor plaster and had corrugated

steel siding as the outside construction and end and side matched sheathing as the inside construction. The leakage at 15 mph was 8.1 cfh per square foot. Had paper been applied vertically between the studding and the boards as in Panel 4*A*, the leakage would have been less than 1 cu ft per hour per square foot of wall.

INFILTRATION THROUGH SINGLE-SURFACED WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

Fig. 25 shows the results of tests on panels having a single thickness. These

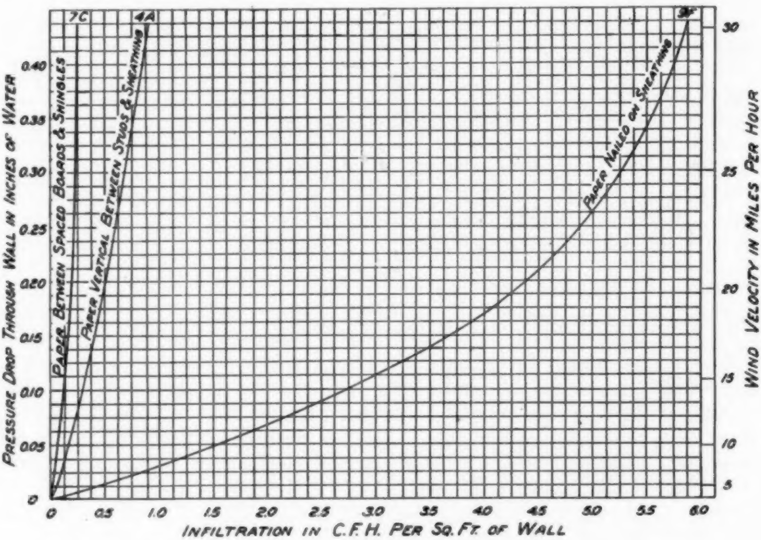

FIG. 18. STUDY OF VARIOUS APPLICATIONS OF SHEATHING PAPER

include panels having tongued and grooved boards both butt ended and end matched, boards and battens and corrugated steel siding.

The largest leakage occurred with Panel 6*D* which was built of 1x10 in. boards spaced on 11 in. centers and the spaces covered with 1x4 in. boards. This leakage was 86.9 cfh per square foot of wall at 15 mph. Fig. 13 is a photograph of this panel.

The leakage through the three panels 4*F*, 9*E* and 10*C* of air dried end and side matched sheathing of a uniform material averaged 35.1 cfh per square foot of wall at 15 mph. The inclusion of the fourth panel of a different material, Panel 4*A*1, of end and side matched sheathing increased the average to 46.4 cfh per square foot of wall at 15 mph. Air dried side matched sheathing in Panels 2*A* and 3*A* showed an average leakage of 12.3 cfh per square foot at 15 mph.

Two panels, 6*B* and 10*E*, were tested having corrugated steel siding. The

leakage through Panel 6*B* was 45.4 cu ft and through Panel 10*E*, 9.1 cfh per square foot at 15 mph. The average of the two is 27.3 cfh per square foot. The large variation in results is explainable in a difference in the length and fit of

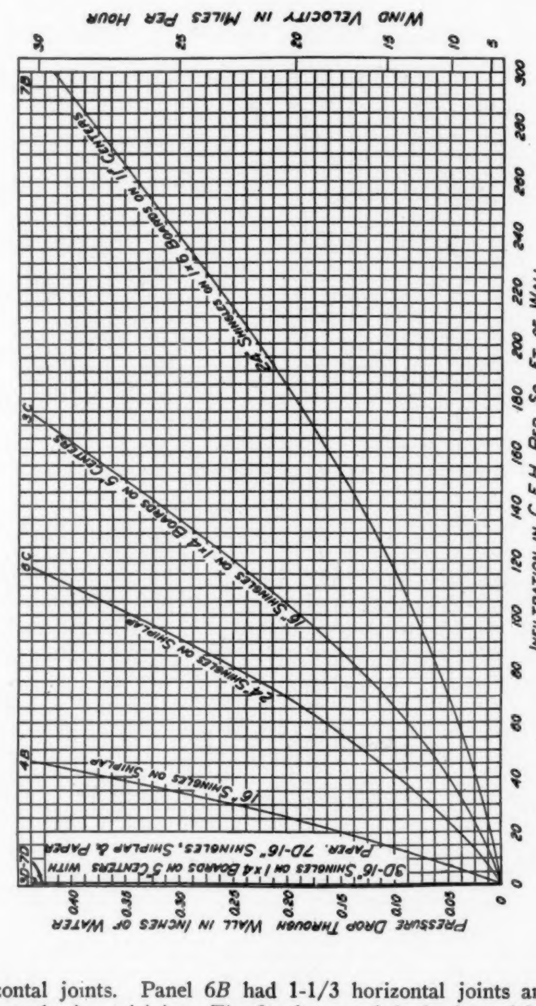

FIG. 19. INFILTRATION THROUGH VARIOUS TYPES OF SHINGLE CONSTRUCTION

the horizontal joints. Panel 6*B* had 1-1/3 horizontal joints and Panel 10*E* had only one horizontal joint. The fit of some of the horizontal joints on Panel 6*B* was poorer than that on Panel 10*E*.

Fig. 12 shows a photograph of Panel 6*B*. The joint between the steel siding

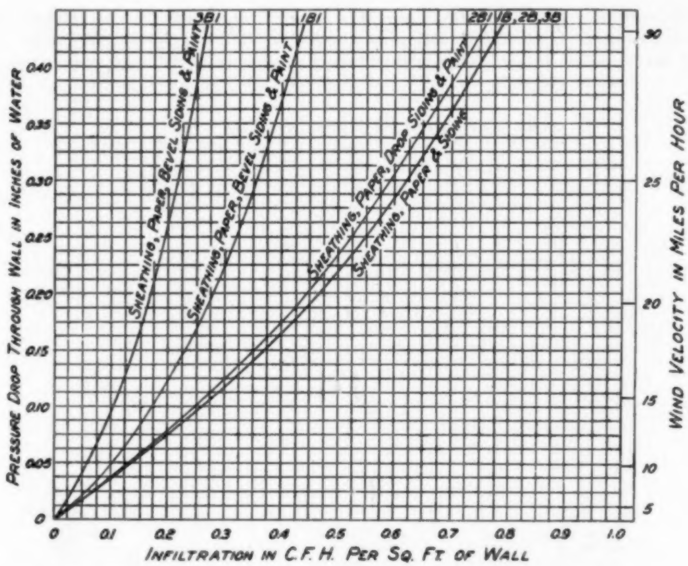

FIG. 20. ADDITION OF PAINT TO DROP SIDING AND BEVEL SIDING

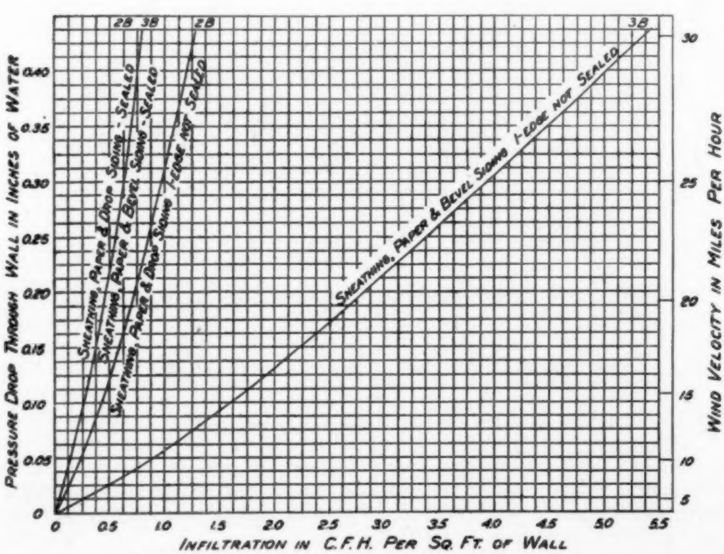

FIG. 21. STUDY OF GOOD AND POOR CONSTRUCTIONS

and the test frame was calked on all four sides during the tests. Had air been allowed to enter the ends of the corrugations at the top or bottom of the wall, the leakage would have been exceedingly large. Such would be the condition in actual building construction unless special care were taken in the application to butt the sheets against wood construction.

The tests on all of the single-surfaced walls show considerable leakage. The application of sheathing paper nailed vertically on the studding to any single-surfaced wall would greatly reduce the leakage. The leakage of such an appli-

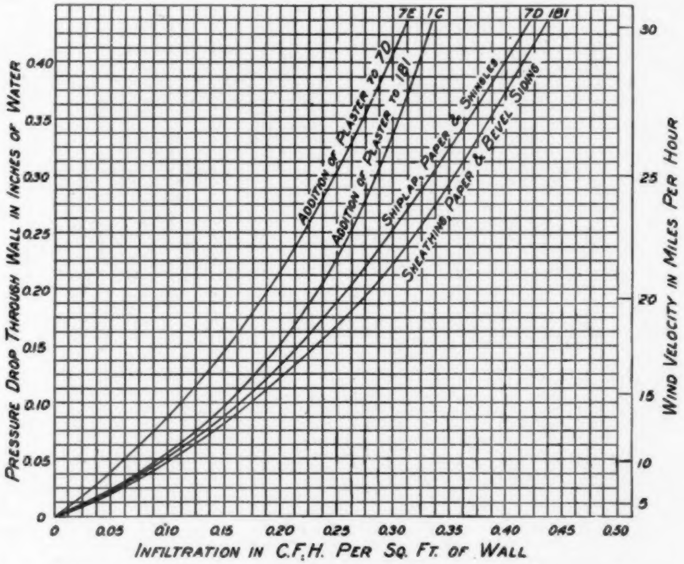

FIG. 22. ADDITION OF PLASTER TO WALLS HAVING SHEATHING PAPER

cation was shown in the test of Panel 4*A* to be less than 1 cu ft per hour per square foot of wall at 15 mph.

CONCLUSIONS

The air infiltration through a frame wall construction containing building paper or plaster properly applied is negligibly small.

The best application of building paper consists of the clamping of the paper between two thicknesses such as sheathing and drop siding or shingles. In a single-surfaced wall, the paper may be effectively applied vertically on the studding under the boards. Special precautions are necessary at the corner of a building or against door or window openings to make the paper effective.

While no difference has been found in the value of a high and a low grade paper on these infiltration tests, likely the good grade of paper will better maintain its efficiency against air infiltration over a period of years than will

the poor grade of paper. A paper having considerable weight and strength should develop fewer defects in application than would a poor paper. The use of a poor grade of paper is particularly objectionable for a construction in which the paper is not firmly clamped between two thicknesses of material over its entire surface.

End and side matched sheathing has a somewhat greater leakage to air than has side matched sheathing. The difference is of no importance when sheathing paper is included in the construction. End and side matched sheathing presents

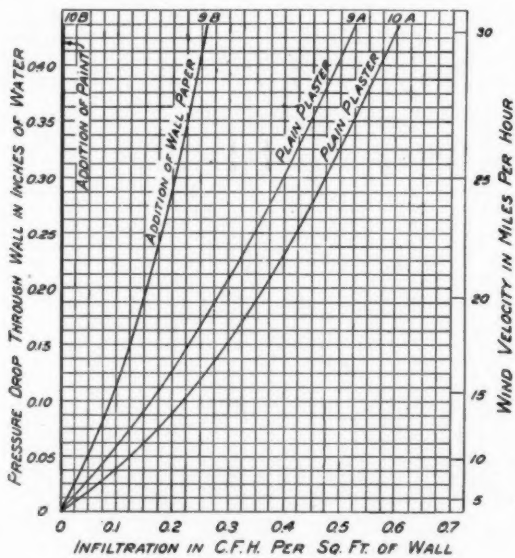

FIG. 23. ADDITION OF WALL PAPER AND PAINT TO PLASTER

the same desirable flat surface for application of paper as does side matched sheathing. There is no disadvantage from the infiltration standpoint to offset the advantages of the use of end and side matched sheathing on the usual building construction. With end and side matched sheathing, random lengths may be used, resulting in a saving in material and sawing labor.

The application of building paper to single-surfaced frame walls such as are used for farm and other shelter buildings can be effectively and cheaply performed. The application of the paper vertically between the studding and the sheathing and with the laps on the studding makes a very good construction from the infiltration standpoint.

To obtain the full efficiency of a frame wall against air leakage, special care must be used at the corners of a building and against window and door openings. Building paper should be carefully wrapped around the corner so as to prevent air getting through the wall construction at this point.

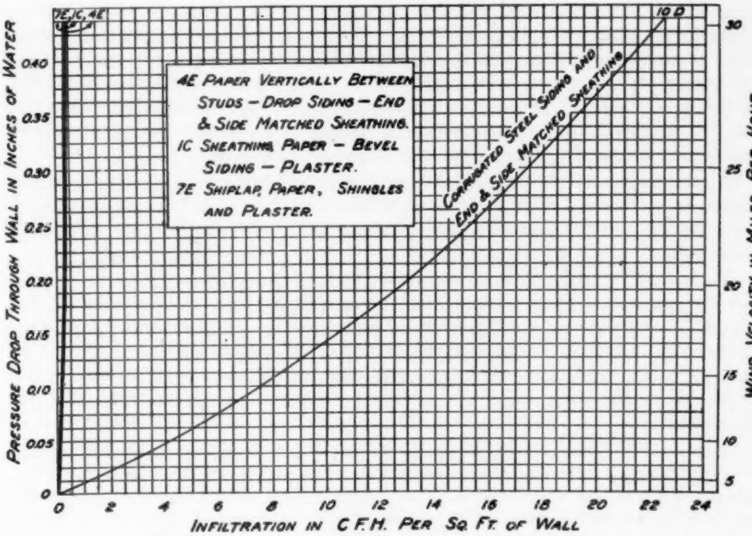

FIG. 24. INFILTRATION THROUGH VARIOUS TYPES OF COMPOUND WALLS

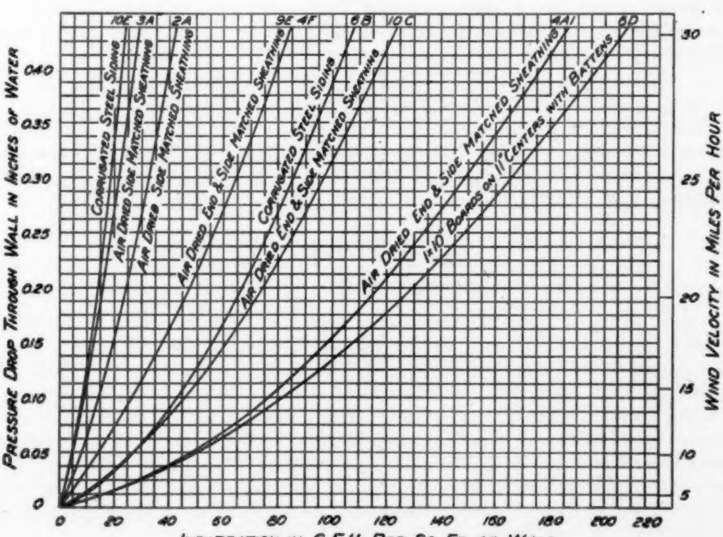

FIG. 25. INFILTRATION THROUGH SINGLE SURFACE WALLS USED IN FARM AND OTHER SHELTER BUILDINGS

In the case of bevel siding, there is the chance for air to enter at a corner construction and travel horizontally to seek out defects in the sheathing paper. This requires the careful application of the paper so as to prevent defects developing and the use of wide laps so as to bring them under the contact lines of the bevel siding. The use of a good grade of paper would be economical due to freedom from tearing during application and the maintaining of a high degree of effectiveness throughout the life of the building.

Drop siding does not present spaces for air travel over the surface of the sheathing paper and very effectively holds the paper against the sheathing. Poor corner construction is a less serious factor with drop siding than with bevel siding.

The use of building paper is justified from an infiltration standpoint in any shingle roof or side wall construction. The use of shiplap rather than spaced boards is desirable in the clamping of the sheathing paper securely over its entire surface. This minimizes the buckling of the paper from aging. In districts where heating seasons are mild, the inclusion of building paper and the use of shiplap rather than spaced boards would be justified by the reduction in infiltration.

Plaster by itself allows the passage of only a negligible amount of air when properly applied. Proper application means the sealing at the baseboard and against window and door openings by running the plaster tightly against the floor or frame construction of an opening. The full effectiveness of plaster is probably seldom obtained. The development of cracks reduces the efficiency of the plaster. The application of paint or wall paper is not justified from the standpoint of infiltration reduction on a plaster wall that is fully effective, but on a cracked plaster surface the application of either paint or paper may reduce infiltration considerably.

The effectiveness of plaster properly applied is no justification for the use of low grade building paper or of the poor construction of the wall containing it. Not only is it difficult to secure and maintain the full effectiveness of the plaster but also it is highly desirable to have two points of high resistance to air flow with an air space between them to keep heat transmission losses at a minimum.

APPENDIX

Description of Test Panels

Items 1 *and* 2. Eight test panel frames were built of 4x8 timbers as described in the forepart of the paper. To the top and bottom members of each of these frames single 2x4 plates were nailed 1½ in. from the face of the frame and extending from end to end of the frame opening. Vertical 2x4 studs were installed between these plates on 16-in. centers, except that the end studs were spaced such as to bring end studs tightly against the vertical frame members. Studs were firmly toe nailed to plates and end studs to frame. A recess was made against the frame members in the 2x4 plates and end studs on both sides of the frame into which calking compound was packed to prevent air leakage at this joint.

Item 3. Over the stud frame in panel frame No. 1, 1x6-in. dressed and side matched sheathing was applied with ends butted over studs, fitted with ordinary

care to secure tightness and face nailed twice at each stud with 8*d* nails. Joints were broken over studs in every third course. This material was green and of No. 1 Common grade. Green sheathing, in this test, consisted of air-dried sheathing that had been soaked in water for several days. The panel was allowed to season 29 days before testing.

Item 4. Over the studding in panel frames Nos. 2 and 3, 1x6-in. dressed and side matched sheathing was applied. Workmanship was in all respects similar to that of *Item* 3, except that the sheathing was thoroughly air-dried when applied. These panels were allowed to season 28 days before testing.

Item 5. Over the stud frame in panel frame No. 4, a good grade of sheathing paper termed, Building Paper *A,* was applied in vertical strips, lapped 2 in. over studs and tacked to studs with roofing nails. Over the building paper, 1x6-in. dressed, and end and side matched No. 1 Common sheathing was applied. Workmanship was similar in all respects to that required for *Item* 3, except that no effort was made to have the end joints between boards occur over studs. End joints were made in not less than two-thirds of the courses, but each board spanned at least two studs. The panel was allowed to season 29 days before testing. Immediately after testing the paper and siding, the paper was cut out between studs with a knife and the panel re-tested.

Item 6. Over the sheathing on panel frames Nos. 2 and 3 after test, Building Paper *A* was applied in horizontal strips lapped not less than 2 in. and with one vertical joint one strip wide, also lapped, in the center of the panel.

Over the building paper on panel No. 2, 1x6 in. drop siding was applied. This siding was air dry to the satisfaction of the inspector, and was nailed in accordance with usual practice. The panel was allowed to season 14 days and then tested. Three coats of good commercial white lead and zinc paint were then applied to the siding at customary intervals, and 6 days after the application of the final coat, the panel was re-tested.

Over the building paper on panel frame No. 3, 1x6-in. bevel siding was applied lapped not less than 1 in. and nailed in accordance with usual practice. The panel was allowed to season 14 days and then tested. Three coats of good commercial white lead and zinc paint were then applied to the siding at customary intervals, and 6 days after the application of the final coat the panel was re-tested.

Item 7. Over the stud frame in panel frame No. 6, Insulation *A* sheathing board was applied, using one vertical joint butted over studding and one horizontal joint clear across panel. The panel was allowed to season 30 days before testing.

Item 8. Over the stud frame in panel frame No. 7, Insulation *B* standard sheathing board was applied in a manner similar to the Insulation *A* (see Item 7) nailed according to manufacturer's specifications. The panel was also allowed to season 30 days before testing.

Item 9. Insulation *A* sheathing board was removed from studding in panel frame No. 6 and 20-gage galvanized corrugated sheet steel siding applied. Sheets were about 27 in. wide and 60 in. long and were lapped and nailed on to the studding according to manufacturer's specifications for siding for farm or industrial buildings. No intentional standing period was allowed for the corrugated steel siding, but 14 days elapsed between the date of construction and the date of test.

Item 10. Over the studding in panel frame No. 9, wood lath and gypsum plaster were applied in accordance with the specifications given in the next paragraph. Lath was No. 1 grade, soft pine, spaced at least ¼ in. apart and nailed to each stud with a 3*d* 16-gage wire nail. Joints were broken at every seventh course. Lath was thoroughly soaked before application and also well wetted down several hours before plastering. Wood grounds ⅞ in. thick were nailed around the stud frame and flush with the outside edges of the plates and the two outermost studs. The crevices between grounds and the panel frame were carefully calked with plasted calking compound. A seasoning period of 15 days was allowed before testing.

"Plaster shall be a good three-coat job of gypsum plaster applied as follows: Scratch coat shall be one part plaster, hair fibered, to not more than two parts (by weight) of dry sand. Brown coat shall be of similar materials and proportions, but unfibered. The finish coat shall be four parts gypsum plaster, one part finishing lime, and five parts clean dry sand."

Item 11. After the plaster in Item 10 on Panel No. 9 was tested, a good grade of wall paper was applied. This was allowed to dry for 10 days and the panel re-tested.

Item 12. Over the studding in panel frame No. 10, metal lath and gypsum plaster were applied in accordance with the following specifications: "The metal lath shall be expanded metal lath weighing not less than 2.5 lb per square yard, attached to each stud with 6*d* nails spaced not over 6 in. apart. Clinch of nails to be upward. Lath to be lapped at sides not less than ½ in. with lower sheet over the upper. Grounds shall be ⅞ in. as for wood lath and plaster. There shall be at least one vertical and one horizontal lap in the metal lath joint entirely across the panel.

"Gypsum plaster shall be used and applied in three coats, as follows: Scratch coat shall be one part plaster, hair fibered, to not more than two parts (by weight) of dry sand. Scratch coat shall be applied with sufficient pressure to fill all meshes and obtain good key.

"Brown coat shall be one part plaster, unfibered, to not more than two parts (by weight) of dry sand. This coat shall be kept back sufficiently from grounds to allow for finishing coat and surface shall be roughened to receive finishing coat.

"The finish coat shall be four parts gypsum plaster, one part finishing lime and five parts clean dry sand."

Item 13. Over the metal lath and plaster on panel frame No. 10 (see *Item* 12) a coat of sizing and two coats of a good commercial flat wall paint were applied. A drying period of 10 days was allowed, after which the panel was re-tested.

Item 14. The wood lath and plaster were removed from panel frame No. 9 and the stud faces carefully cleaned. Over the stud frame Insulation *A* plaster lath was applied in commercial sheets; the sheets being broken over studs twice vertically and once horizontally. Lath was nailed according to manufacturer's specifications. Grounds were ⅞ in. as for wood lath and plaster. A scratch coat and finish coat of stucco were then applied.

Item 15. The bevel siding and building paper on panel frame No. 1 were removed. Over the matched sheathing, expanded metal stucco reinforcement of

20-gage thickness with openings ¾ in. by 2 in., and weighing 1.8 lb per square yard, was applied. Stucco reinforcement was placed horizontally and fastened to the sheathing with saddle nails spaced about 8 in. apart over the surface. Vertical laps were made over studs and horizontal laps laced with wire. Stucco was applied according to the following specifications:

"Scratch coat shall be of one part portland cement to three parts sand, applied ½ in. thick and troweled well through the reinforcement. It shall be thoroughly dry before the brown coat is applied. The brown coat shall be ½ in. thick, of the same proportions, and the finish coat from ⅛ in. to ¼ in. thick. Total thickness of stucco over sheathing shall be approximately 1⅛ in. and shall be established by 1⅛ in. grounds nailed around the outside of the stud frame." The panel was allowed to season 29 days before testing.

Item 16. The bevel siding, building paper and sheathing were removed from panel frame No. 3. Over the stud frame 1x4-in. butt-edged boards No. 2 common grade were applied spaced 5 in. on centers. Over the boards, edge grain 5/2-16-in. red cedar shingles were applied 5 in. to the weather using zinc coated shingle nails. The panel stood 69 days before testing.

Item 17. The sheathing and building paper on panel frame No. 4 were removed. Over the stud frame, 1x8-in. shiplap No. 2 common was applied butted over studs and face-nailed twice. Over the shiplap edge grain 5/2-16 in. red cedar shingles were applied 5 in. to the weather using zinc coated shingle nails. The panel stood 14 days before testing.

Item 18. The Insulation *B* standard sheathing board was removed from panel frame No. 7. Over the studs, 1x6-in. butt edged boards, No. 2 common grade were applied spaced 11 in. on centers. Over the boards, edge grain strictly clear 24-in. red cedar shingles were applied, 11 in. to weather. The panel was allowed to stand 15 days before testing.

Item 19. The corrugated sheet metal was removed from panel frame No. 6. Over the studding, 1x8 in. shiplap No. 2 common grade was applied. Over shiplap, 24-in. edge grain red cedar shingles, strictly clear, were applied, 11 in. to weather. The standing period was 14 days before testing.

Item 20. The 24-in. shingles were removed from panel frame No. 7. All remaining nails were pulled or driven flat to avoid any projections on the boards. Building Paper *B*, a good grade of paper, was then applied over boards according to manufacturer's instructions. Over the paper, strictly clear edge grain 24-in. red cedar shingles were applied 11 in. to the weather, using zinc coated nails. The panel was allowed to stand 67 days before testing.

Item 21. The 16-in. shingles were removed from panel frame No. 3. All remaining shingle nails were pulled or driven flat to avoid any projections on the boards. Building Paper *B* was then applied over the boards according to manufacturer's instructions. Over the paper, edge grain 5/2-16 in. red cedar shingles were applied 5 in. to the weather using zinc coated shingle nails. The panel stood 69 days before testing.

Item 22. The matched sheathing was removed from panel frame No. 1 and butt-edged 1x6-in. sheathing, No. 1 Common grade, was applied over studs. The lumber was air dry and fitted with ordinary care. Over the sheathing, Building Paper *B* was applied according to manufacturer's specifications. Over the building paper, 1x10 red cedar bevel siding, *B* grade and better, was applied,

lapped 1½-in., and double nailed with 8*d* cement-coated box nails. The panel stood for 70 days before testing.

Two coats of good quality white paint were then applied to the bevel siding at the customary intervals, and 6 days after application of the second coat the panel was re-tested.

Item 23. On the reverse side of panel frame No. 1, finished as specified in *Item* 22, wood lath and gypsum plaster were applied in conformity to specifications for *Item* 10. The panel stood 68 days and was again re-tested.

Item 24. All previous construction was removed from the 2x4 studs in panel frame No. 9. Over studs air dry 1x6 No. 1 Common end and side matched sheathing was applied. The panel was allowed to stand 2 days and then tested.

Item 25. Over the sheathing in *Item* 24 ordinary red resin-sized building paper was applied with laps horizontal, nailed with No. 3 shingle nails spaced approximately 7 in. The panel was re-tested immediately.

Item 26. Over the building paper mentioned in *Item* 25, air dry 1x6 drop siding was applied, nailed in accordance with usual practice, and painted with three coats of good commercial white lead and zinc paint applied at customary intervals. The panel was allowed to season 13 days and again re-tested.

Item 27. All previous construction was removed from the 2x4 studs in panel frame No. 10. Over the studs 1x6 end and side matched sheathing was applied. The panel was allowed to season for 7 days and then tested.

Item 28. Over the other side of panel frame No. 10 corrugated steel siding was applied as in *Item* 9. The panel was re-tested four days later, the steel siding being exposed to the pressure side of test machine.

Item 29. All previous construction was removed from the 2x4 studs in panel frame No. 7. Over the studs 1x6 air dry shiplap, double nailed, Building Paper *B*, and 16 in. red cedar shingles 5/2 thickness were applied 7½ in. to the weather. The panel was allowed to season 7 days and then tested.

Item 30. Over the other side of panel frame No. 7, wood lath and plaster were applied according to the specifications of *Item* 10. The panel was allowed to season 14 days and re-tested.

Item 31. All previous construction was removed from the 2x4 studs in panel frame No. 4. Over the 2x4 studs, Building Paper *B* lapped vertical on studs, and 1x6 air dry Select drop siding were applied. Siding was painted with two coats of commercial white lead and zinc paint. After the application of final coat of paint, the panel was allowed to dry for 4 days and then tested.

Item 32. Over the other side of panel frame No. 4, 1x6 air dry end and side matched sheathing was applied. Panel was allowed to season 6 days and then re-tested.

Item 33. Drop siding and building paper (*Item* 31) were removed from panel frame No. 4 and the panel immediately re-tested with the sheathing in *Item* 32 exposed to the pressure side of the test machine.

Item 34. All previous construction was removed from panel frame No. 6, including inside vertical studs, leaving only the 2x4 plates nailed to top and bottom frame members and the two end studs nailed to vertical frame mem-

bers. Between the two end studs and perpendicular to them two 2x4 girts were toe-nailed on 33-in. centers. Over the studs and girts and perpendicular to the latter, 1x10 boards were applied with a one inch space between boards nailed with 8*d* wire nails at each side and in the middle of the boards at each girt. One heavy coat of good commercial paint was then applied. Immediately after painting, 1x4 battens were applied over the one inch spaces between the 1x10 boards and nailed on one side only of the battens through one edge of the 10-in. boards into the girts but not at other points. After the first coat of paint had dried, an additional coat was applied to boards and battens. The panel was allowed to season 14 days out-of-doors before testing.

DISCUSSION

A. P. KRATZ: This paper, taken in conjunction with the previous reports on the same subject, emphasizes the importance of proper wall construction. It brings out the fact that the major part of air leakage results from improper or faulty construction, and that the proper use of building paper, plaster, and paint, reduces such leakage to a negligible amount.

Since it is impossible to standardize faulty construction, it is difficult to choose, in spite of a wealth of information, just what factors should be used in the case of a particular building, and the infiltration loss remains one of the most uncertain calculations in determining the predicted heat loss from a building. It would seem to the writer, from the evidence presented, that the next step in a future program should be a study of the actual infiltration in existing buildings in order to establish upper and lower limits for the factors to be used for given types of constructions in conjunction with the factors now established for known conditions.

D. R. BREWSTER: These tests emphasize the importance of building paper to such an extent that I thought it might be of interest in this connection to call attention to a study of building papers made by the Bureau of Standards, the results of which are recorded in a bulletin entitled A Study of Sheathing Papers, Research Paper No. 85, of the Bureau of Standards, published in the *Journal of Research,* Vol. III, July 1929. This can be bought separately.

The conclusions of the study made by the Bureau of Standards bear out the results reported in this paper to an interesting degree and point out that the importance of sheathing paper depends not so much upon its weight and thickness and cost as upon the method of application, the tightness of the joints and the care used in preventing tearing and damage to the paper in use. Thus the only justification for using a high grade paper in building construction is to provide a paper of sufficient strength so that in application it is not apt to tear and so that during the life of the building it will not be subject to deterioration from the influence of moisture and rain blowing in. It is evident that it really pays to use a high grade building paper from the standpoint of obtaining a maximum length of life of the effectiveness of the construction in preventing infiltration.

Among the conclusions of the U. S. Bureau of Standards study I might read the following sentences which are of interest: "The price range of sheathing papers is about ten-fold between the cheapest and the most expensive. There is no very definite relation between the price of such papers and

their value as building papers." This brings out the importance of the method of application rather than the quality of the paper itself. "Strength sufficient to insure getting the material in place, whole, water resistance sufficient to insure against damage to plastering and inside finish, which sometimes occurs in severe storms, and impermeability to air, are apparently the requisite elements in a good sheathing paper." It is evident that high grade building papers will more nearly meet these requirements than cheap, low grade papers that are made to meet price competition rather than to give definite standards of quality and performance.

No. 869

SURFACE CONDUCTANCES AS AFFECTED BY AIR VELOCITY, TEMPERATURE AND CHARACTER OF SURFACE

By F. B. Rowley,[1] A. B. Algren,[2] and J. L. Blackshaw,[3] Minneapolis, Minn.
MEMBERS

The results of cooperative research between the University of Minnesota and the American Society of Heating and Ventilating Engineers

IN MAKING a complete analysis of the heat flow through built-up wall sections, that is, the flow of heat from air on one side of a wall to air on the other side, three points must be considered: *first,* the flow of heat from the air on one side of the wall to the surface on the same side, which is assumed to be the same as the flow of heat from the opposite surface to the air on its side; *second,* the conductance of the heat through those parts of the wall that are built up of homogeneous materials; *third,* the conductance of heat across air spaces within the wall. These air spaces may be of any size or shape, and there may be any number of them in the wall structure. If the laws governing these three factors are definitely known, then it is possible to calculate the heat flow through any built-up wall section, provided the characteristics of the materials used in the wall are also known.

The present investigation is part of the co-operative research program between the American Society of Heating and Ventilating Engineers and the *University of Minnesota.* The object of the investigation was to determine the effect of air velocity, temperature, and surface characteristics on heat transmission from surfaces. Surface conductance (f) is defined as the number of British thermal units which will flow between one square foot of the surface of the material and the surrounding air per hour per degree difference in temperature between the surface and the air. A preliminary report of this investigation was published in the December 1929, A.S.H.V.E. Journal. Since that time, the apparatus has been remodeled and the work has been extended to cover nine different surfaces used in building construction. In order to make this discussion clear, it will be necessary to use two of the line drawings shown

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Minneapolis, Minn., June, 1930.

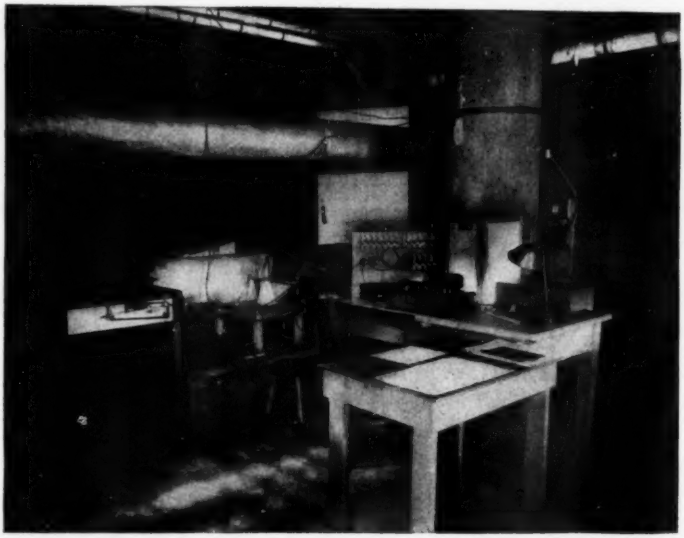

FIG. 1. FRONT VIEW OF SURFACE CONDUCTANCE TEST APPARATUS

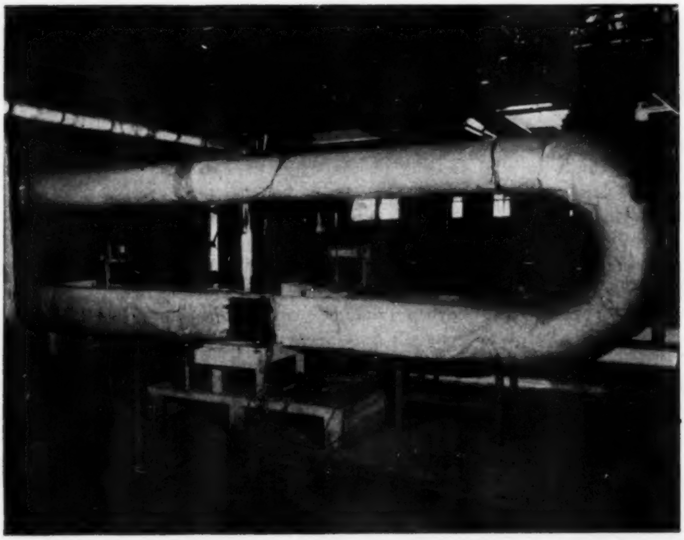

FIG. 2. REAR VIEW OF SURFACE CONDUCTANCE TEST APPARATUS

in the former paper, because these drawings represent the fundamental principles of the apparatus.

One point which should be definitely understood is that these investigations cover air velocities *parallel* to the surface. The question immediately arises as to what the condition would be if the air flow were *perpendicular* or at some other angle to the surface. This is a point for further investigation; the relative surface effect of wind to different angles of the surface must be determined and factors obtained which can be applied to the various surface coefficients to correct for direction of air flow.

It was decided in this investigation to determine the coefficients with the air passing parallel to the surface. Therefore, it was necessary to construct ap-

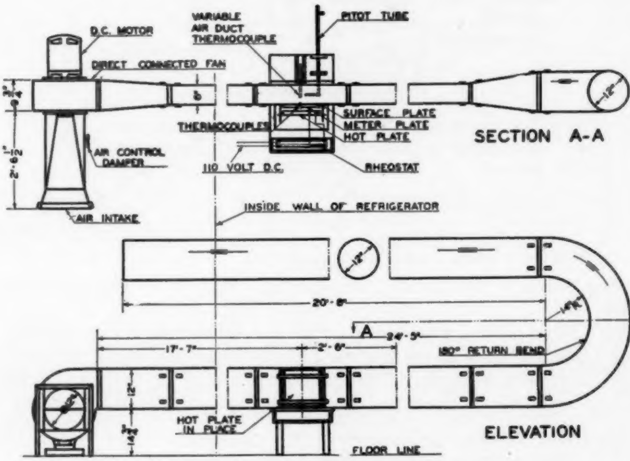

FIG. 3. PLAN AND ELEVATION OF TEST APPARATUS FOR DETERMINING SURFACE CONDUCTANCES

paratus which would provide: *first,* air moving over a test surface at various constant velocities; *second,* accurately controlled air temperatures; *third,* test surfaces which could be supplied with a measured amount of heat; *fourth,* instruments for measuring the air velocities and temperatures of the air over the test surfaces and the amount of heat flowing through the test surface. In order to obtain these conditions, the apparatus was set up as shown in the photographs, Figs. 1 and 2, and the line drawings, Figs. 3 and 4. Air of the proper temperature was supplied by a large refrigerator capable of maintaining temperatures down to —10F. This air was blown by a 12-in. multi-blade fan, driven by a ½ hp, 850 rpm, 220 volt, d-c, direct-connected, variable speed motor. The air from the fan passed through a long, straight, 6-in. x 12-in. rectangular duct in order to eliminate eddy currents. Seventeen feet from the fan, the air passed over the 12 in. square test surface which was inserted in the side of the duct flush with its inside surface. After this, it was carried through a return bend

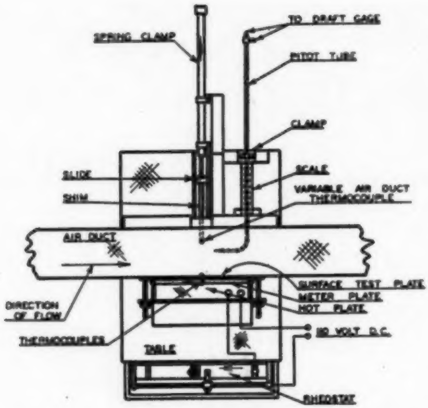

FIG. 4. PLAN VIEW SHOWING ARRANGEMENT OF METER PLATE, TEST SPECIMEN, THERMOCOUPLE AND PITOT TUBE IN RELATION TO AIR DUCT

and brought back to the cold room. This arrangement provided a flow of air past the test surface without turbulence.

The arrangement of the test surface, together with the method of measuring the air flow and the air temperatures, can be best understood by referring to Fig. 4 which is an enlarged section to Fig. 3. As shown in this drawing, the test material was placed with the test surface flush with the inside surface of the air duct. The test material varied in thickness from ¼ in. to ½ in., depending upon the type of surface. Heat for the test surface was applied by a hot plate,

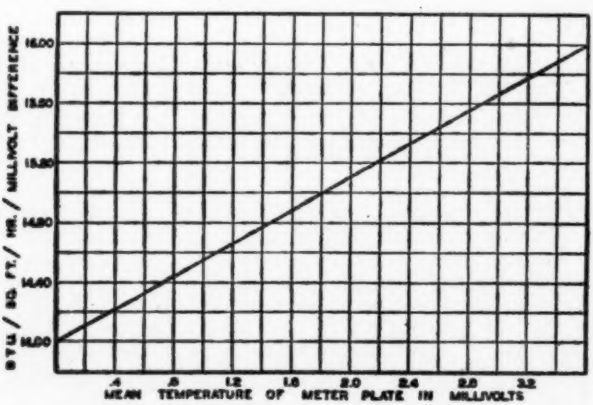

FIG. 5. CALIBRATION CURVE FOR HEAT FLOW METER

the quantity of heat being measured by passing it through a heat meter. The hot plate was electrically heated with 110-volt direct current, rheostat controlled, and checked with an ammeter and a voltmeter to obtain uniform heating conditions.

The meter plate, constructed of ⅛ in. Bakelite, was substantially the same as the Nicholls heat flow meter. Two parallel series of 28 pairs of 28-gage copper-constantan junctions, differentially wound on the plate, were used. Of these, one series of 56 couples served to check the other series. Surface temperatures of the meter plate were also taken with the aid of three 28-gage,

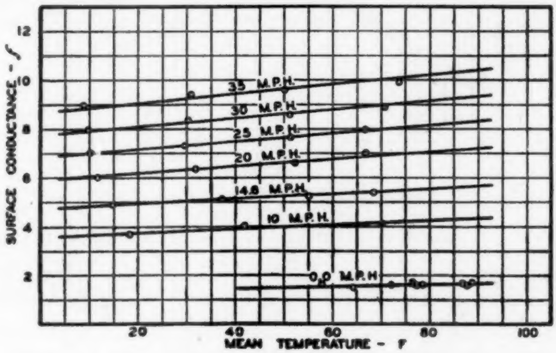

FIG. 6. CONSTANT VELOCITY CURVES FOR GLASS SURFACE

copper-constantan thermocouples on either side. This meter plate was directly calibrated on the hot plate test apparatus at the *University of Minnesota*. FIG. 5 shows the calibration curve for this meter. The average of the hot and cold side thermocouple readings gave the mean temperature of the meter directly in millivolts. The curve gives the corresponding number of Btu which will flow per square foot per hour per millivolt of the differential couple series. A simple multiplication of this value with the differential couple reading gives directly the number of Btu flowing through a square foot of the plate per hour. This meter plate is many times more sensitive than was the older plate used in preparing the previous report. It also has a greater conductance and, therefore, allows much more heat to flow from the test surface, thus giving a more accurate determination of the surface conductance. Two such meters were constructed, and when placed in series on the same test, their readings checked within 0.17 per cent.

The air velocity at varying distances from the test surface was measured by means of a Pitot tube and draft gage. For the lower velocities, the Wahlen gage was used, but to higher velocities, the inclined draft gage was found satisfactory. The air velocity was derived from the formula.

$$V = 12.456 \times \sqrt{\frac{P_v}{w}}$$

where V = velocity of air in miles per hour

P_v = inches of water indicated as the velocity pressure by the Pitot tube

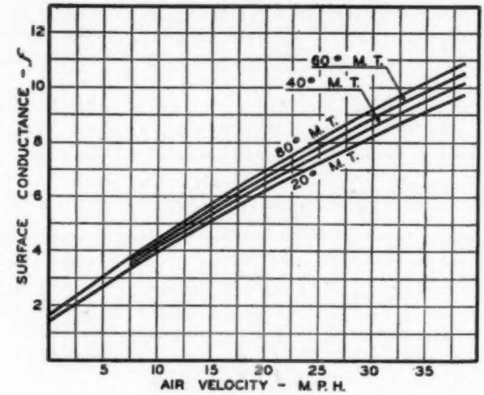

FIG. 7. CONSTANT MEAN TEMPERATURE CURVES FOR GLASS SURFACE

w = weight of air as obtained from tables and charts with the aid of wet- and dry-bulb thermometer readings, together with the barometric reading.

Air temperatures were measured by a 24-gage, copper-constantan search thermocouple mounted on a carriage arranged so that it could be moved in and out from the test surface and held at any predetermined distance. Tests were made both with a shield between the thermocouple and the test surface, and

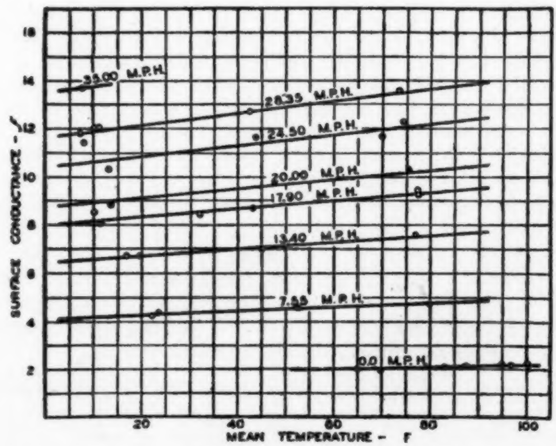

FIG. 8. CONSTANT VELOCITY CURVES FOR BRICK SURFACE

without it. It was found that, at the comparatively low temperatures used, radiation had no effect on the temperatures indicated by the couple. Therefore, most of the tests were made without this shield. All thermocouple readings were read on a potentiometer; the cold junction in all cases was an ice-bath.

It will readily be seen that with this set-up varying air velocities could be obtained over the test section. The velocity and temperature of the air could be accurately measured at any distance from the surface. The air temperatures were kept constant by holding a constant temperature in the refrigerating room. This was done by setting the refrigerator to give air at a slightly lower temperature than that required. The air was brought up to the required temperature by a thermostatically controlled electric heating element placed within the room.

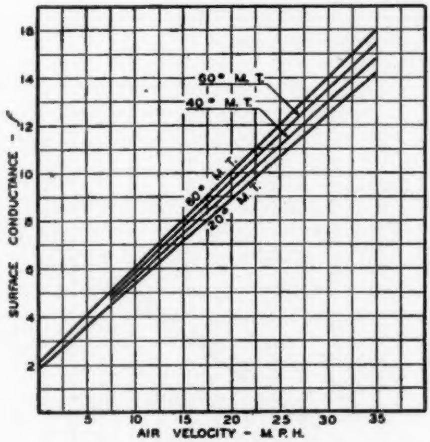

FIG. 9. CONSTANT MEAN TEMPERATURE CURVES FOR BRICK SURFACE

By this means, very constant temperatures could be maintained throughout any length of test period.

In order to obtain average radiation conditions, the inside surface of the test duct was painted a dull gray, and all of the pipe outside the refrigerator was covered with a one-inch thick blanket insulating material. With this arrangement, the surfaces immediately around the test surface were at substantially the same temperature as the surrounding air, practically the same as that for the average wall. The insulation also prevented condensation on the outside of the air duct when low temperature air was used.

In determining the surface temperatures, two different methods were tried. *First,* the couple was embedded to bring the junction flush with the test surface; *second,* it was rigidly attached to the surface and covered with a thin vellum paper. When the thermocouple was embedded, the difference between the air temperature and the indicated surface temperature was greater than when it was placed on the surface, and, therefore, the calculated surface coefficients were

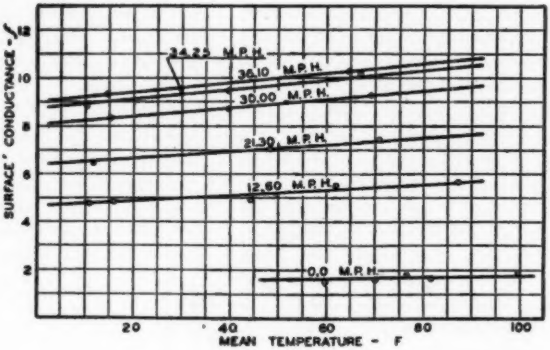

FIG. 10. CONSTANT VELOCITY CURVES FOR WHITE PAINT SURFACE ON PINE

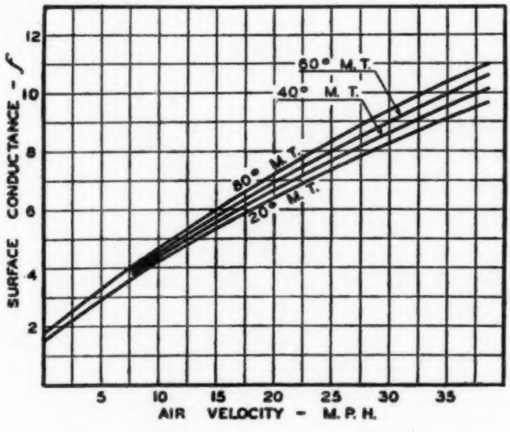

FIG. 11. CONSTANT MEAN TEMPERATURE CURVE FOR WHITE PAINT SURFACE ON PINE

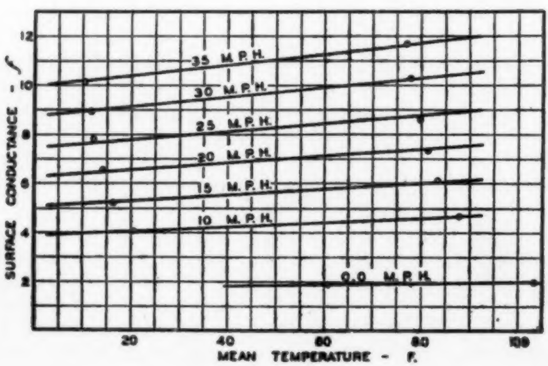

FIG. 12. CONSTANT VELOCITY CURVES FOR SMOOTH PLASTER SURFACE

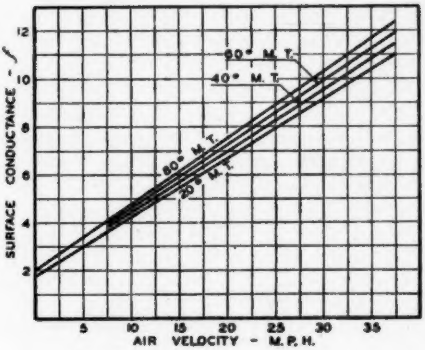

FIG. 13. CONSTANT MEAN TEMPERATURE CURVES FOR SMOOTH PLASTER SURFACE

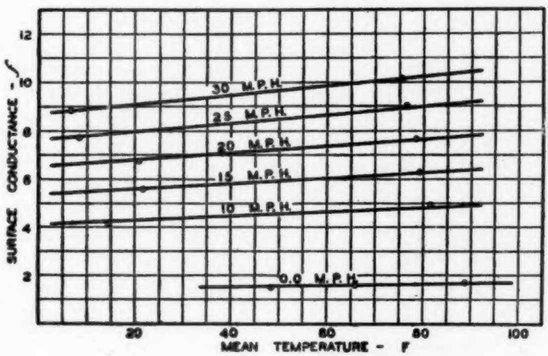

FIG. 14. CONSTANT VELOCITY CURVES FOR CLEAR PINE SURFACE

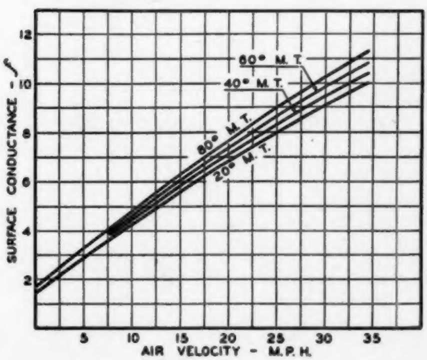

FIG. 15. CONSTANT MEAN TEMPERATURE CURVES FOR CLEAR PINE SURFACE

TABLE 1. TEST DATA FOR SMOOTH PLASTER SURFACE

Item No.	Test No.	Air Velocity mph	Meter Plate Temperatures Millivolts				Air and Surface Temperatures Degrees Fahrenheit				Heat Flow Btu Sq Ft Hr	Surface Conductance f
			Differential	Hot Side of Meter	Cold Side of Meter	M. T. of Meter	Test Surface	Air. Temp. 1 In. From Test Surface	M. T.	Temp. Difference		
1.......	252	35.00	10.324	1.018	0.629	0.823	17.55	2.85	10.20	14.70	149.233	10.152
2.......	251	30.00	10.538	1.090	0.696	0.893	20.20	3.05	11.62	17.15	152.748	8.906
3.......	253	25.00	10.402	1.102	0.715	0.908	21.75	2.45	12.10	19.30	150.829	7.815
4.......	254	20.00	10.507	1.202	0.803	1.002	25.70	2.35	14.02	23.35	152.929	6.549
5.......	255	15.00	10.504	1.302	0.907	1.104	31.00	1.55	16.27	29.45	153.463	5.211
6.......	256	10.00	10.384	1.440	1.053	1.246	39.25	1.90	20.57	37.35	152.540	4.084
7.......	257	0.0	6.584	2.008	1.759	1.883	86.45	34.85	60.65	51.60	99.089	1.920
8.......	258	35.00	9.746	2.363	1.993	2.178	83.25	70.60	76.92	12.65	148.285	11.722
9.......	259	30.00	9.790	2.399	2.028	2.213	84.95	70.50	77.72	14.45	149.150	10.322
10.......	260	25.00	9.852	2.474	2.103	2.288	88.45	71.05	79.75	17.40	150.489	8.648
11.......	261	20.00	9.795	2.524	2.157	2.340	91.50	71.25	81.38	20.25	149.863	7.400
12.......	262	15.00	9.777	2.600	2.245	2.422	95.60	71.30	83.45	24.30	150.028	6.173
13.......	263	10.00	9.550	2.735	2.392	2.563	103.40	72.30	87.85	31.10	147.308	4.737
14.......	264	0.0	5.970	2.823	2.603	2.713	126.15	80.35	103.25	45.80	92.594	2.022

somewhat lower. This difference was not great, however, and seemed to be due to the fact that couples were affected by the lower and warmer material below the surface. The second method was finally selected. The surface couples were made with 28-gage copper-constantan wire flattened out at the junction, thus giving a very thin couple at the point of contact.

In the assembly of the test apparatus, the test surface, together with the heat meter and the hot plate, were placed in the side of the air duct and clamped in place with specially designed clamping screws. The conditions of air velocity, air temperature, and surface temperature were then selected, and the apparatus was operated under these conditions for a sufficient length of time to assure uniform results. The air velocity for the test was measured at the center of the

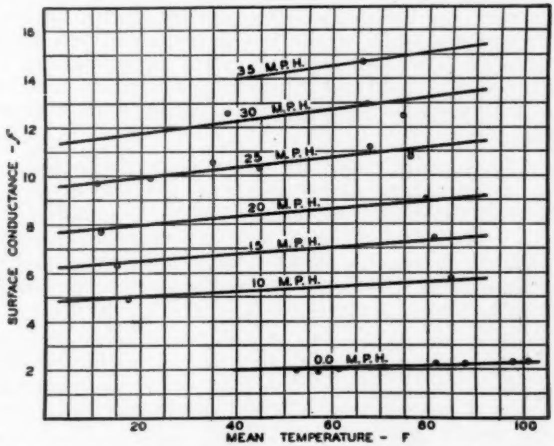

FIG. 16. CONSTANT VELOCITY CURVES FOR CONCRETE SURFACE

duct, because this would be the maximum air velocity over the test surface. The air temperature was measured by placing the search thermocouple one inch from the test surface. Preliminary tests had shown that when the thermocouple was placed in contact with the test surface and gradually moved away from it, the temperature steadily dropped until the couple was about ½-in. from the surface, after which this temperature remained uniform and equal to the air temperature, regardless of distance. One inch distance was therefore taken to be reasonable, and was maintained throughout all tests.

The obtaining of an accurate test necessitated the holding of constant conditions over a considerable length of time. Data for a test were not taken until preliminary observations, taken at fifteen minute intervals, showed that the heat flow, room, surface, and air temperatures, and air velocity were constant. These readings took three or four hours between tests. A test consisted of recording the following data:

1. Type of surface.
2. Test number, date, time, name of observer.

3. Barometric reading, wet- and dry-bulb temperatures.
4. Voltage on blower-fan and inlet damper-setting.
5. If Wahlen gage were used, air temperature to determine the specific gravity of the alcohol in the gage.
6. Pitot tube readings at distances varying from 0.125 in. to 3 in. from the test surface.
7. Differential, hot, and cold meter plate readings.
8. Hot plate voltage and amperage.
9. Surface temperature.
10. Air temperatures at distances from the surface varying from 0.1 in. to 3 in.

Because all temperatures were taken at or near the center of the meter plate and the test surface, and because the meter plate and the test surface were very thin, no allowance was made for end loss of heat from these plates. As an additional precaution, the edges of these plates were insulated with heavy layers of felt to stop any heat loss. Mean temperatures for a test were taken as the average between the air temperature and the surface temperature of the test specimen.

Values of the surface conductance for each test were obtained by dividing the total heat leaving the test surface per square foot per hour by the temperature difference between the surface and the air one inch away from the surface. Values of the surface conductance of any one surface vary with the mean temperature and the air velocity. Keeping one of these two variables constant, a series of tests were made to determine the effect of the other and the results were plotted in the form of a curve.

Surfaces which were considered to be most typical of building construction were used for these tests. In making tests on each surface, several different air velocities were selected and runs were made at different mean temperatures for each velocity. The following surfaces were tested: glass, brick, white paint on pine, smooth plaster, clear white pine, rough plaster, concrete, and stucco. Very complete tests were made on the first three of these surfaces, four or more points being taken for each respective air velocity at different mean temperatures ranging from zero to 100 F. It was found that these points lay practically on a straight line and that when they were plotted on a large scale graph and the lines were extended, these lines crossed the line of zero surface conductance at absolute zero mean temperature, or, in other words, that with total absence of heat, the surface conductance was zero. This was true of all surfaces tested.

As representative of a typical test, the complete data taken for smooth plaster are shown in Table 1. Final results for all of the tests are shown in Figs. 6 to 22, inclusive.

Figs. 6 and 7 give data obtained on the glass surface, tests having been made at air velocities ranging from zero to 35 mph. Fig. 6 shows the relation between surface conductance and mean temperature at certain constant velocities. The constant mean temperature curves for glass, Fig. 7, were derived from the curves shown in Fig. 6. Reading up the 20 F mean temperature line of Fig. 6, points representing surface conductances were read on each of the air velocity curves. These points were then plotted against air velocity on the

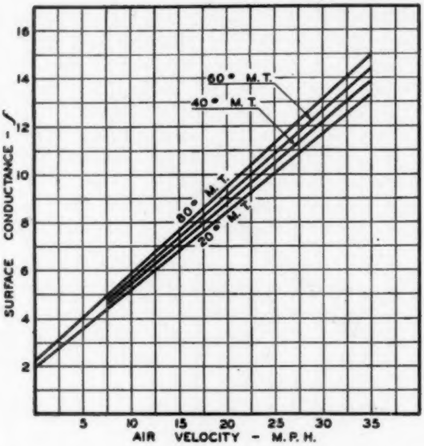

FIG. 17. CONSTANT MEAN TEMPERATURE CURVES FOR CONCRETE SURFACE

graph shown in Fig. 7 to give the 20 F mean temperature curve. The 40 F, 60 F, and 80 F curves were obtained in like manner. Other mean temperature curves might have been obtained for any limits within the curve of Fig. 6, or for limits outside of the curve by extending the constant velocity lines. Other values can be interpolated directly from the curve sheet, Fig. 7. Eddy currents and turbulent air flow prevented tests from being taken at velocities greater

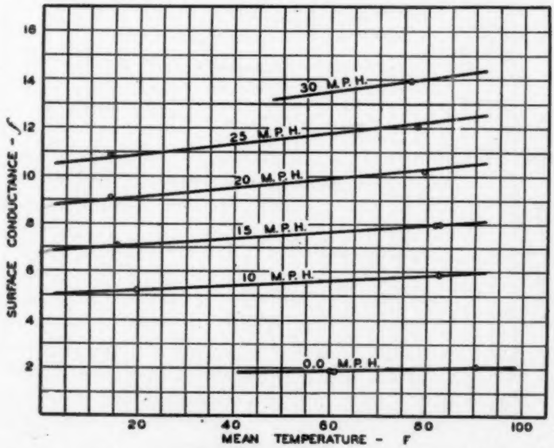

FIG. 18. CONSTANT VELOCITY CURVES FOR ROUGH PLASTER SURFACE

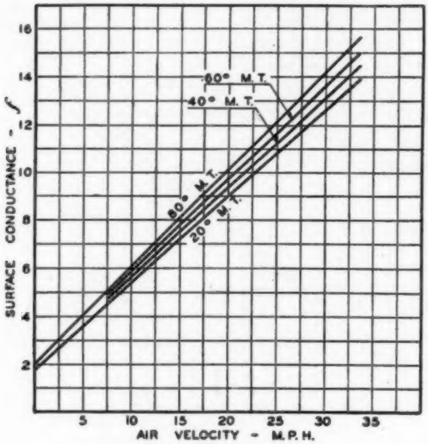

FIG. 19. CONSTANT MEAN TEMPERATURE CURVES FOR ROUGH PLASTER SURFACE

than 35 mph. Rough surfaces caused more difficulty at high velocities than smooth ones.

The curves in Figs. 8 to 21, inclusive, need no specific explanation. They represent different surfaces tested and are similar to the curves in Figs. 6 and 7.

The curves in Fig. 22 show the surface conductance for each of the materials

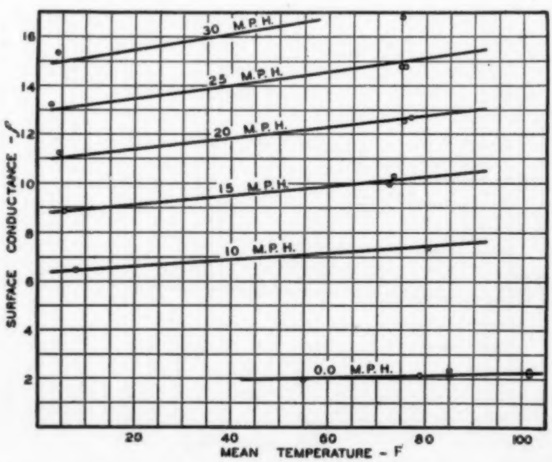

FIG. 20. CONSTANT VELOCITY CURVES FOR STUCCO SURFACE

tested at a mean temperature of 20 F. This mean temperature was selected as representative of average wall conditions. From this group of curves, it is evident that materials may be placed in groups according to their surface characteristics. The curves for glass and white paint on pine coincide, and curves for similar surfaces would, no doubt, fall close to this line. The curves for clear pine and smooth plaster are substantially the same, and it is assumed that curves of other smooth wood surfaces would correspond to them. Brick and rough plaster show identical results, and the concrete surface curve lies but slightly below them probably because the concrete surface was smoother than either the brick or plaster. The stucco surface was very rough and irregular.

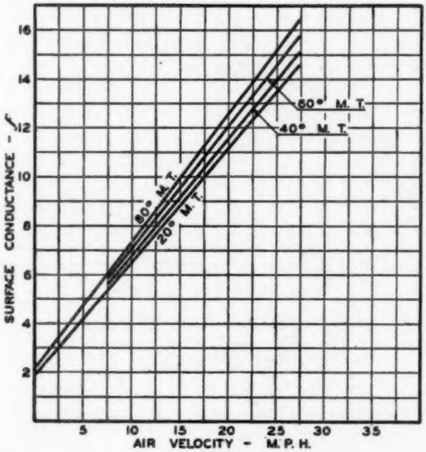

FIG. 21. CONSTANT MEAN TEMPERATURE CURVES FOR STUCCO SURFACE

It is difficult to determine the coefficients for such surfaces, and there would doubtless be a variation depending upon the surface irregularity. Tests were also made on a sand-coated surface painted white, but no curve sheets are included for these tests. Results of this surface, if plotted on Fig. 22, would have placed a curve slightly below that for concrete. For all surfaces tested, the surface conductances at zero air velocity fall between 1.4 and 2.0. While these values were recorded as zero velocities, it is understood that zero velocity is impossible to attain with vertical surfaces because of convection currents.

Whether or not humidity affects surface conductance is a point to be considered. If it does, the effect is of small consequence because, although there was no provision made for keeping constant humidities in these tests, there was no appreciable variance in the results. The effect of a wet surface on the surface coefficients is also a question for further consideration.

It is significant that the results of the tests showing surface coefficients at zero

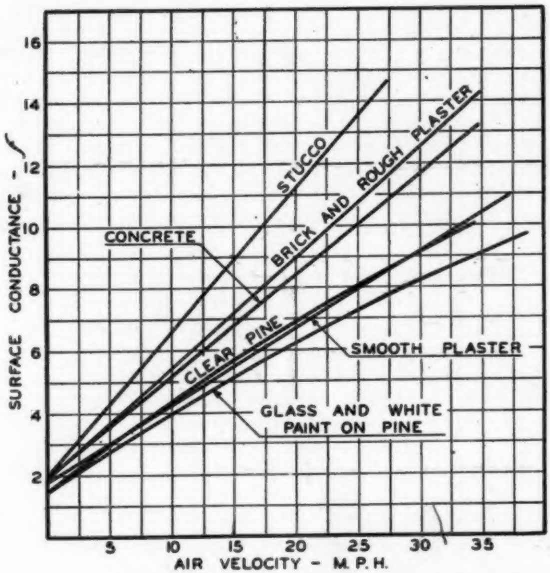

FIG. 22. CURVES SHOWING RELATION BETWEEN SURFACE CONDUCTANCES FOR DIFFERENT SURFACES AT A MEAN TEMPERATURE OF 20F

velocity check closely with those which were obtained in former tests on similar materials, using the hot box set-up for built-up wall section tests at the *University of Minnesota,* which set-up does not allow for any movement of air.

DISCUSSION

R. E. BACKSTROM (WRITTEN): Fig. 22 is of principal interest since it compares the coefficients of various types of surfaces to each other. These curves show that, with a wind velocity of 15 mph, which is commonly accepted as an average velocity to use in calculating the overall heat transmission factors of wall sections, the conductance of all types of surfaces is considerably more than the value now used and given in THE GUIDE as 4.02. Rough surfaces, particularly, show higher conductances, and even glass, a very smooth surface, shows a conductance of about 5.0 At zero air velocity the smooth surfaces of glass, clear pine and painted pine have a conductance of 1.4, approximately the same as used in THE GUIDE, while the surfaces of smooth plaster, concrete, brick, rough plaster and stucco have higher values.

Applying this correction to the test results of Wall No. 8 as reported in the paper Overall Heat Transmission Coefficients Obtained by Tests and by

Calculation, by F. B. Rowley, A. B. Algren and J. L. Blackshaw, and presented at the 1929 summer meeting of the Society, the coefficient of a frame wall having lap siding, paper and wood sheathing as outside construction and smooth plaster on wood lath inside, is found to be increased from 0.226 in still air to 0.251 with a wind velocity of 15 mph. This value compares with 0.262, the value reported in THE GUIDE 1930.

With these comprehensive data on surface conductance, and with the data that these authors have previously submitted on the conductances of air spaces and various building and insulating materials, it is now possible to compute with accuracy the overall transmission coefficients for built-up sections. The fact that these data are available for different mean temperatures should not be overlooked in selecting the proper value to use in making the calculations.

D. R. BREWSTER (WRITTEN): FIG. 22 is particularly interesting and valuable because of the clearcut relationship brought out between the conductance of materials with a rough surface and those with a smooth surface. One is led to wonder why it is that a rough surface in the form of stucco has a conductance at the higher velocities nearly twice that of smooth surfaces such as glass or painted wood. This might be a worth-while subject for further research.

It occurs to me that this greater conductance may be due to a considerably larger area of exposed surface per superficial square foot of material just as mountains have much more exposed surface per square mile than flat country. It may be possible to work out a direct ratio between these factors if some method could be devised for measuring the full amount of exposed surface on a rough material.

I note that the velocity measurements were taken in the middle of the 12-in. flue. I am wondering to what extent the air in actual contact with the surface was slowed down by surface friction and whether this friction at the edge of the stream of air should be taken into account in a research such as this or whether it may be ignored.

The value of new data such as these on surface conductance is very apparent in the building industries and indicates that a succession of dead air spaces in wall or roof construction may have a composite insulating value equal to if not greater than much heavier and more expensive solid construction, particularly if infiltration is properly taken care of.

A. P. KRATZ AND A. C. WILLARD (WRITTEN): This paper presents results on an important aspect of heat transmission, concerning which we have had entirely too little actual data. Many years ago we attempted to measure surface conductances at various air velocities over 10 or 12 different wall materials. The results of our work as presented in University of Illinois Engineering Experiment Station *Bulletin No. 102* show very satisfactory agreement with Professor Rowley's values for still air, ranging from 1.10 to 2.00 Btu for still air as compared with 1.4 to 2.00 Btu in the present paper.

We did not get the uniform increase in surface conductance with wind velocity which Professor Rowley reports in his paper for moving air. Our values increased with wind movement quite rapidly at first, but the rate of increase with air motion became less and less as the wind velocity rose. It

seems to us that some such diminishing rate of increase in surface conductance is to be expected with increasing air motion in the case of large areas such as exist in actual buildings. If such reasoning is valid, the fact that we used much larger test surfaces than the one square foot test surface used by Professor Rowley may serve to explain the discrepancy between his surface conductances and ours at high wind velocities. For velocities below 15 mph the agreement between his average values for moving air and ours is fairly satisfactory, and justifies our continued confidence in the calculated heat transmission coefficients now published in THE GUIDE.

F. C. HOUGHTEN: A year or two ago it was recognized that there was a need for further data on surface transmission coefficients and air space coefficients. During the past year data have been supplied by the University of Minnesota on both of these subjects and the additional data contained in the curves and tabulations in this paper should be helpful in strengthening our knowledge of heat transfer and our ability to calculate overall transmission coefficients from the various factors that go to make up for the component parts of a wall.

The Laboratory in Pittsburgh is working on another phase of the subject of surface coefficients. In this study, the interest is not so much in determining the coefficients for a large number of surfaces but rather it has been in studying the effects of air velocity and the place with reference to the surface, where the air velocity is measured, on the coefficient. The Laboratory also hopes to study some other factors, including radiation and convection losses from the surface.

For the few tests already made on two surfaces in Pittsburgh the results bear out almost exactly the curve presented today by Professor Rowley. The Laboratory will probably publish something further on the subject in the near future.

The most important question before us now in regard to surface transmission coefficients is not the value of the coefficients for any particular velocity at some particular point near the surface, but rather it is knowledge concerning the velocity near a wall for any given velocity out in the open. For example, when the Weather Bureau station reports a 40-mile wind velocity out in the open, what is the velocity parallel to a wall surface at a distance of an inch or a half inch from that surface as measured by Professor Rowley in his study? That is the particular phase of the subject that the Laboratory is working on.

PRESIDENT HARDING: The reported results appear to indicate higher values than those observed at the University of Illinois in 1914-15. There is perhaps a good reason for this difference as the velocity of the air was measured at different distances from the surface in these two series of tests. I believe the test data from the two tests will be found comparable when Director Houghten finishes up his correlation data which it is anticipated he will do this year. We want to know what the velocity is at a certain distance from the surface when the Weather Bureau reports a 15 or 30-mph wind velocity in a free and unobstructed space, so that we may find out whether the present surface coefficients in common use are right or wrong.

No. 870

WALL SURFACE TEMPERATURES

By A. C. WILLARD[1] AND A. P. KRATZ,[2] URBANA, ILLINOIS

MEMBERS

A great deal of attention has been devoted in recent years to the so-called surface coefficients for building walls, and especially to the ratio which exists between the outside and inside surface coefficients, which must be known in order to compute the overall heat transmission coefficient for any given wall.

It so happens that this ratio is always the reciprocal of the ratio of the temperature drops at the surfaces at the inside and outside faces of the wall, and hence may be accurately ascertained in the case of any existing wall by simply measuring the inside and outside air temperatures and the corresponding surface temperatures.

Such measurements have been in progress at the University of Illinois for several years, and a study of the results reveals the fact that an entirely new significance attaches to the inside surface temperature of a building wall, and the difference between this temperature and the air temperature inside the building. The inside wall surface temperature not only affects personal comfort, but may be made use of for estimating the relative thermal values of various walls as actually built.

For example, would you be more comfortable on a zero day in (1) a building with outside walls made entirely of thin sheet iron, or (2) a building with outside walls made entirely of thick corkboard, provided the same "breathing line" temperature of 70 F existed in both buildings?

In other words, the thermal effectiveness of the walls used in a heated or cooled building is reflected in the difference in temperature which that wall maintains between the inside air temperature at the "breathing line" and the inside wall surface temperature at the same level. This paper discusses the new significance of such temperatures and presents actual original data on the subject.

INTRODUCTION

DURING the investigation[3] of various types of heating systems at the *University of Illinois* an increasing amount of attention has been given to the surface temperatures of the exposed walls of heated rooms as actually determined under typical winter weather conditions. In the past, much importance has been attached to the air temperatures at various levels within the heated rooms, and such temperatures, usually taken at the "breath-

[1] Professor of Heating and Ventilation and Head of the Department of Mechanical Engineering, University of Illinois, Urbana, Illinois.

[2] Research Professor in Mechanical Engineering, Engineering Experiment Station, University of Illinois, Urbana, Illinois.

[3] An investigation of warm air furnace heating has been in progress at the *University of Illinois* since October, '918, under a co-operative research agreement between the *National Warm Air Heating Association* and the University. Six Engineering Experiment Station Bulletins and one Circular have already been published.

An investigation of direct systems of steam and hot-water heating has also been in progress since April, 1926, under a similar co-operative research agreement between the *Institute of Boiler and Radiator Manufacturers*, the *Illinois Master Plumbers' Association*, and the University. Two Engineering Experiment Station Bulletins have already been published.

* Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Minneapolis, Minn., June, 1930.

FIG. 1. WARM-AIR HEATING RESEARCH RESIDENCE AT THE UNIVERSITY OF ILLINOIS

ing line" 5 ft above the floor, have been regarded as a measure of heating plant performance. Any plant which maintained 70 F at the breathing line in coldest weather was supposed to satisfy the heating guarantees and the occupant.

The occupant has not always agreed with this idea. In the first place, he occupies a zone at an average height of 2 ft 6 in. above the floor where the air is always colder[4] than at the breathing line. How much colder depends on the type of heating plant and heating unit and house construction, but is not the subject of the present discussion, important though it is. In the second place, but possibly of more importance than the first item, the occupant is also subjected to the radiation effect of the cold surfaces of the exposed walls and glass (assuming floors and ceilings are next to heated spaces), *which are always colder than the air in the room at the same level.*

There is nothing particularly novel about the preceding statement, but the heating engineer appears to have taken very little interest in this fact aside from the academic reference made to it in setting up the general equations for heat flow through any wall. The surface temperature on the inner, or room side, of the exposed wall is usually denoted as (t_1), and, "since it is unknown," is promptly eliminated from the final equation and from the reader's mind. Unfortunately, the occupant in the actual room is keenly conscious of its reality, and the colder the weather and the poorer the wall construction, the more conscious of his cold surroundings he becomes *even though all "breathing line" temperatures are exactly 70 F.*

SUMMARY OF RESULTS

Actual measurements, over a considerable period of time, of both air and wall surface temperatures, at the Research Residence (Fig. 1) and at the room heating testing plant (Fig. 2) in the Laboratory at Urbana, with the breathing line temperature at approximately 70 F in all cases, show:

(1) A rapid increase in the difference between air temperatures at the "breathing line" and the inner wall surface temperatures at the same level, as the outside air temperature drops.

Reference to Fig. 3 will show that when the indoor-outdoor air temperature difference increases from 20 to 70 F, the difference between the temperature of the air at the breathing line and the inner wall surface increases from 2.5 to 12.5 F, although there is no change in the indoor air temperature at the breathing line. If the occupants were just comfortable in the first case when the outdoor air temperature was approximately 50 F, they will not be comfortable in the second case when the outdoor air temperature was approximately 0 F.

(2) For any given outside air temperature, the inside air to surface difference in temperature increases with the outside wind velocity over the wall.

The very marked effect of wind velocity on the "inner air to surface" difference in temperature is strikingly shown in Figs. 4 and 6 which present data from the room heating testing plant in the Laboratory. Note the drop and change in slope of the inside wall surface temperature curve caused by wind (curve No. 3 compared with curve No. 2).

(3) At any outside air temperature, the "inner air to surface" difference

[4] University of Illinois Engineering Experiment Station Bulletins No. 189, pages 60 and 61, 67 and 68, and No. 192, pages 40 to 44.

ILLINOIS ENGINEERING EXPERIMENT STATION

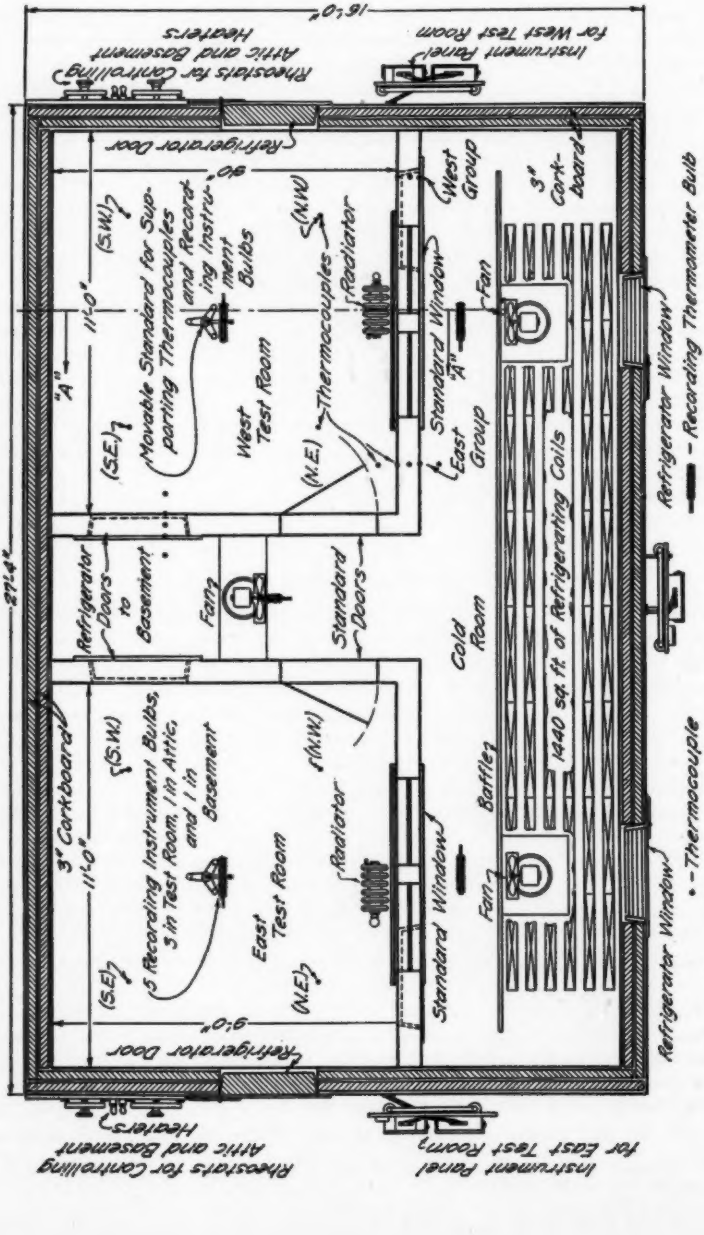

FIG. 2. DIRECT-STEAM AND HOT WATER HEATING LABORATORY PLANT AT THE UNIVERSITY OF ILLINOIS

in temperature is greater than the "outer surface to air" difference in temperature, except for still air conditions.

An inspection of Fig. 3 at any indoor-outdoor air temperature difference shows the temperature difference A is always greater than B.

(4) For a wall exposed to actual weather conditions with wind velocities ranging from 5 to 10 mph, the ratio between the "inner air to surface" difference and the "outer surface to air" difference in temperature increases rapidly as the outside temperatures drop.

This fact has never been recognized or allowed for in heat loss calculations for moderate temperature differences. The decided increase in this ratio occurring at the Research Residence is shown in Fig. 3, from which it may be noted that the "inner air to surface" difference in temperature A is not only greater than the "outer air to surface" difference B, but the ratio A/B also increases from 1.30 to 1.83, with the same wind velocity in all cases. Reference to Fig. 6, however, for the Laboratory plant indicates that at approximately the same wind velocity, the ratio A/B is practically constant at a value of 3.2 over the whole range of outdoor temperatures.

A number of factors are present in the case of the wall at the Research Residence that are not present in the Laboratory plant. The principal difference in the two cases is that the wall in the Research Residence is always under the influence of some solar radiation, or "sky-shine," even on dark and cloudy days. This effect is completely lacking in the case of the wall in the Laboratory plant. An explanation of the difference in the behavior of the two walls can be based on the hypothesis that this difference is caused entirely by solar radiation. If the latter is present, the outside surface of the wall will be warmed and the ratio A/B will be correspondingly reduced. Furthermore, if the radiation is more intense, or the ratio of sunshine to no sunshine is greater on mild days than on colder ones, the ratio A/B will be reduced more in mild weather than in cold. This is exactly the condition shown in Fig. 3.

In the case of the Laboratory plant (Fig. 6), where the sun or solar radiation effect is absent, there is no warming effect to be correlated with the milder temperatures, and a constant value of the ratio A/B over the whole temperature range is to be expected. Since there is no warming effect from solar radiation, it is also evident that the actual value of the ratio will be higher than for the case in which the solar radiation is present. These hypotheses are also supported by the results shown in Fig. 6.

The conditions at the Laboratory plant, with the total absence of solar radiation, are similar to the laboratory conditions under which the ratios commonly used in practice were determined, and it is of some interest to note that the usual assumption of a ratio of approximately 3 to 1 at about 15 mph wind velocity corresponds very closely with the ratio of 3.2 to 1 found in the Laboratory plant. Furthermore, even in the case of the Research Residence, where the solar radiation somewhat modifies the ratio, the value of 1.83 to 1 obtained at 70 F indoor-outdoor difference and 10-mile wind velocity, when corrected to a 15-mile wind velocity, is reasonably close, for all practical purposes, to the ratio of 3 to 1 usually assumed. The fact that such modification occurs, however, should always be taken into consideration where walls exposed to actual weather conditions are concerned.

(5) The ratio between the "inner air to surface" difference and the "outer surface to air" difference in temperature increases rapidly at any given outside air temperature as the outside wind velocity increases. A fact which is recognized and allowed for in heat loss calculations by making the outside surface

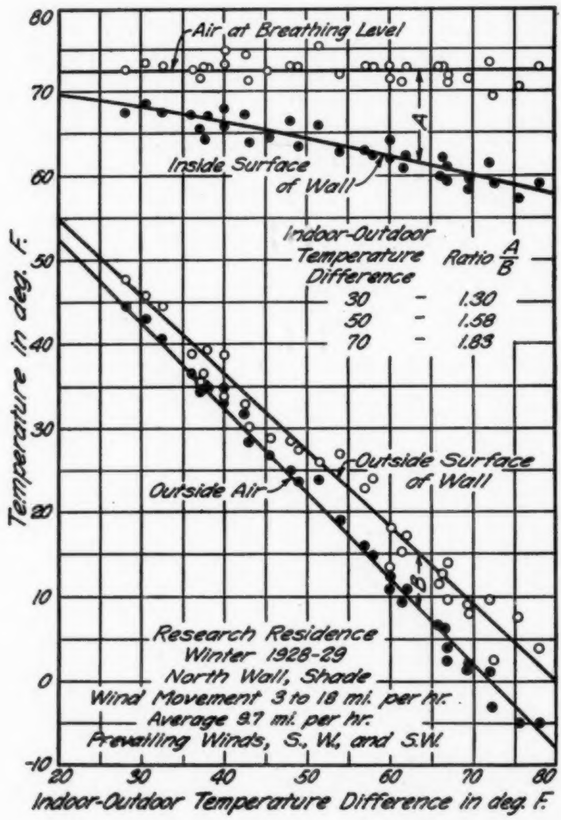

FIG. 3. INSIDE AND OUTSIDE AIR AND WALL SURFACE TEMPERATURES AT RESEARCH RESIDENCE

coefficient greater than the inside surface coefficient. For an inside-outside air temperature difference of 70 F with 15-mile wind velocity, the ratio is ordinarily assumed to be approximately 3:1, ranging from 2.5:1 to 3.5:1. Note especially that the 3:1 ratio does not apply precisely for 15-mile wind velocity at an inside-outside air temperature difference much less than 70 F, since the ratio is affected by the inside-outside air temperature difference as well as the outside wind velocity.

Reference to Figs. 4 and 6 from the Laboratory plant will show that the ratio of "inner air to surface" temperature difference to "outer surface to air" temperature difference increases from approximately 11.6/11.5=1.00 for still air to 17.5/5.5=3.18 for a wind velocity of approximately 10 mph.

Attention is again directed to the absence of solar radiation or "sky-shine" in this plant which operates to reduce somewhat the temperature of the outside

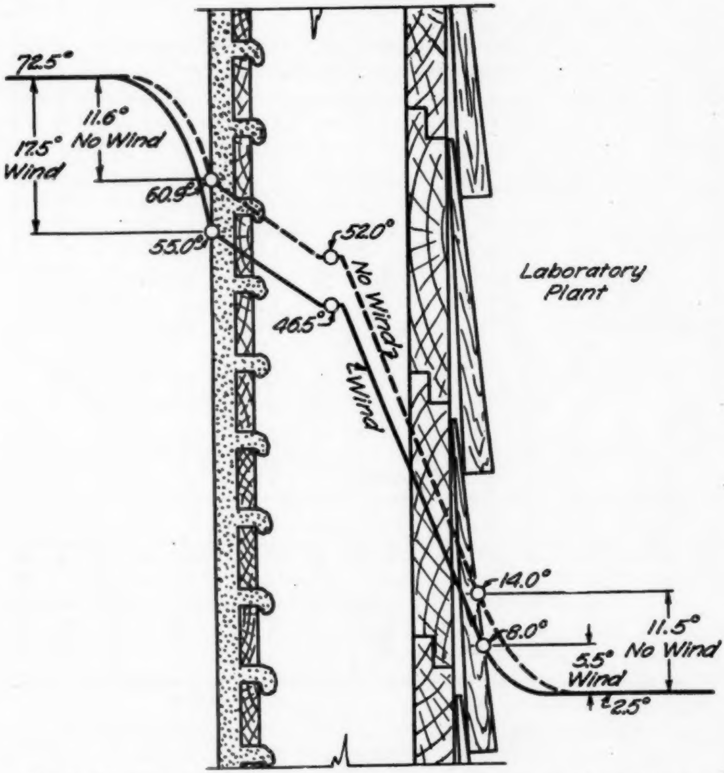

FIG. 4. TEMPERATURE GRADIENTS THROUGH WALL SECTION OF LABORATORY PLANT

wall surface below that for an actual house, and hence the ratio of 3.18 at 10-mile wind velocity is larger than for the same wind velocity at the Research Residence.

(6) Walls exposed to sun effect, especially south walls, show marked reduction in "inner air to surface" differences in temperature (improving personal comfort) for many hours during, and for some time after periods of sunshine. See the period from 7:30 a. m. to 10:00 p. m. on January 7, 1929,

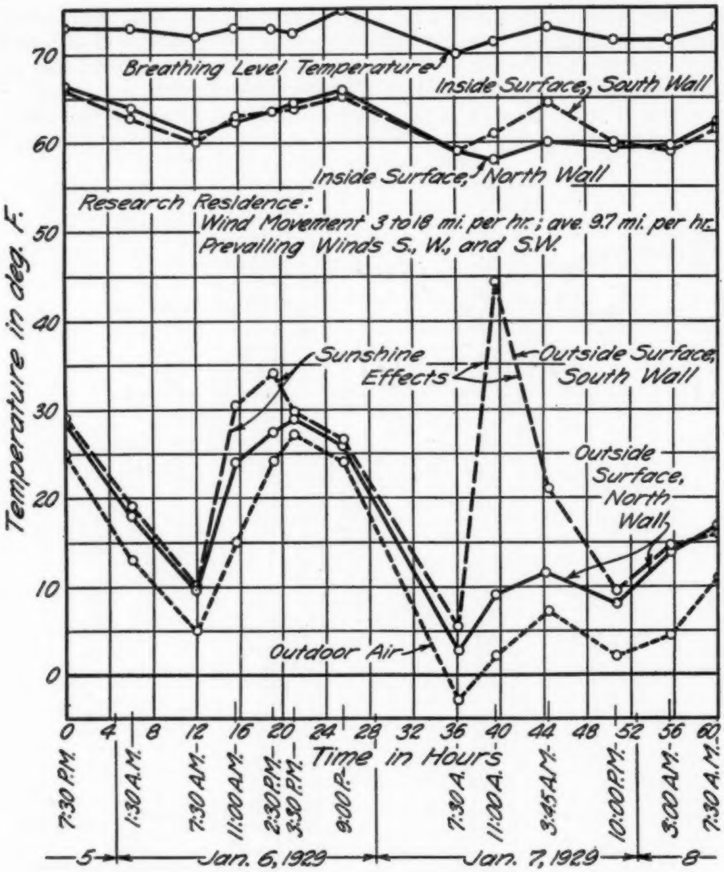

FIG. 5. GRAPHIC LOG OF AIR AND WALL SURFACE TEMPERATURES AT RESEARCH RESIDENCE

Fig. 5, and note that the sun effect lasted long after sunset and darkness, although the outside air temperatures ranged only from 3 F to + 7 F during this period.

The "outside wall to air" difference in temperature show tremendous *increases* for the same south wall during periods of sunshine, ranging from a difference of 8 F at 7:30 a. m. to 43 F at 11:00 a. m. on January 7, 1929, Fig. 5.

(7) On cloudy days, there is apparently very little distinction between the "inner air to surface" difference in temperature of north and south walls. The

same statement applies almost equally well to the "outer surface to air" difference in temperature. See Fig. 5, excepting the period from 7:30 a. m. to 3:30 p. m. on January 6, and the period from 7:30 a. m. to 10:00 p. m. on January 7, 1929, when sun effect was more or less active.

It is noteworthy, however, that the outside surface temperature of the south wall was always slightly higher than the outside surface temperature of the north wall even with no sunshine and at night.

CONCLUSIONS AND A SUGGESTED APPLICATION

(1) Entirely too little attention has been given to the radiation effect of the inside surfaces of exposed wall and glass on the personal comfort of the occupants of heated rooms, especially during "coldest weather."

(2) The use of air temperatures at the "breathing line" alone, as the sole index of satisfactory thermal conditions for room occupancy, is crude and may result in much dissatisfaction on the part of the occupants, since they are subjected to much colder air temperatures at the "comfort line" 2 ft 6 in. above the floor, and also to the radiation effect of much colder exposed wall and glass surfaces at all levels in the room.

(3) The present ratio of 3:1 for the outside to inside surface coefficients used in the academic calculations of heat transmission coefficients of solid walls and glass is only valid for an inside air to outside air temperature difference of approximately 70 F and a wind velocity of approximately 15 mph. *Both air temperature difference* and wind velocity affect this ratio.

(4) Unquestionably, a distinction should be made between north and south rooms having exactly the same *calculated* heat loss, and the same wall and glass exposure, in selecting the sizes of heating units or capacities for such rooms. The distinction should be accomplished by an *addition* to the heating equipment of the north room, and not by a reduction for the south room.

Temperature control or regulation, preferably automatic, is far more important in south rooms than in north rooms, as attempts to control an entire building from the main unit *cannot be successful on sunshiny days.*

(5) The effective insulation of buildings assumes an entirely new significance when the differences between the inside air and inside wall or glass surface temperatures for various types of wall construction and glazing are considered and compared. Effective wall and glass insulation accomplishes two things: (1) it *reduces* heat loss, which is obvious, but from the standpoint of the comfort of the occupants it may render a far more important service in that, (2) it *increases* the inside wall or glass surface temperatures very materially in severe winter weather and thereby reduces the radiation or chilling effect of the cold wall and glass surfaces on the occupant *for the same breathing line air temperature in the room in both cases.*

(6) *The Application.*—Comparisons of the effectiveness of various types of insulation applied to building walls and glass, or of two or more uninsulated walls *after the buildings are erected and occupied and under heat can be made in the field,* by merely measuring the air and wall surface temperatures inside and outside the building. It is desirable to make these temperature measurements either at night or on a cloudy day. Under no circumstances should they

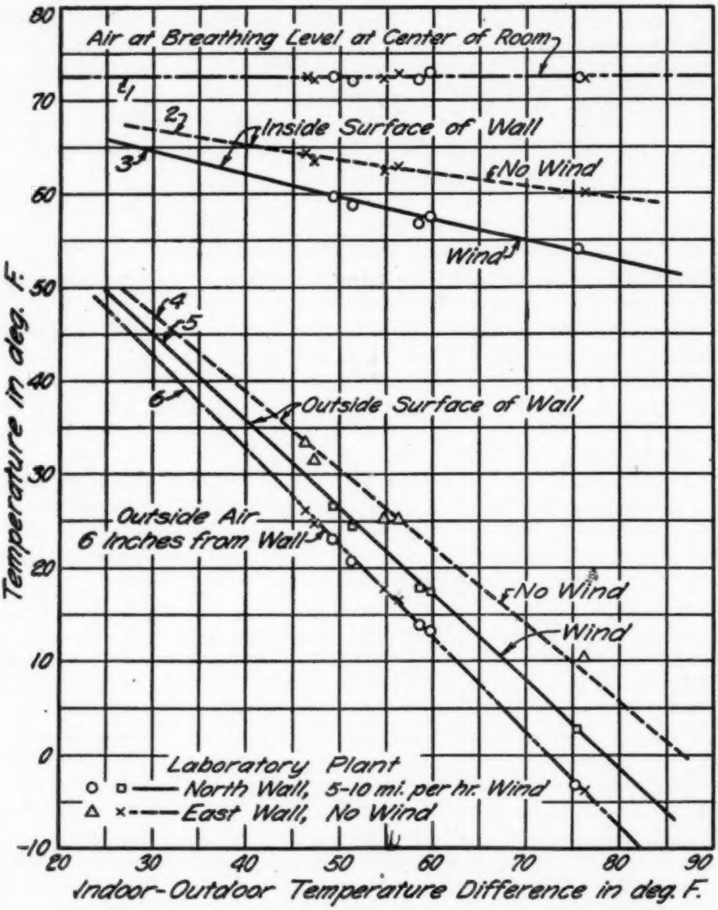

FIG. 6. INSIDE AND OUTSIDE AIR AND WALL SURFACE TEMPERATURES AT LABORATORY PLANT

be made on any wall upon which the sun is or has been shining with any intensity whatever.

By means of a portable thermocouple outfit[5] for reading surface temperatures (one inside and one outside) and the corresponding air temperatures, it is possible to check up the heat transmission and most certainly give any actual wall an exact rating in terms of some standard wall. If the measured

[5] Under *no circumstances* should the attempt be made to read these surface temperatures with ordinary mercury in glass thermometers attached to the surfaces. The authors have found that the resulting errors are relatively, very large.

difference between the inner wall surface temperature and the inside air temperature is less than for some standard wall, for the same inside and outside air temperatures (taken from a chart similar to Fig. 3) then it is a better wall, and a scale of values for walls and glass can easily be set up.

Moreover, the predicted performance of any wall may be checked against its actual performance as finally constructed, by comparing the actual measured surface temperatures with the calculated surface temperatures for the same wall. The calculated[6] surface temperatures are readily found for any given inside and outside air temperatures and wind movement such as may exist on the day the air and surface temperatures were actually measured. For example:

Assume two walls, No. 1 (insulated) has an overall heat transmission coefficient, $U_1=0.12$ Btu per sq ft per hour per degree, and No. 2 (uninsulated) has a value of $U_2=0.24$Btu. Surface coefficients will be the same in both cases, $f_{in}=1.34$ and $f_{out}=3 \times f_{in}=4.02$ for the same wind movement (taken at 15 miles in both cases) and air temperatures taken at, $t=70$ F and $t_o=0$ F.

Wall No. 1

$$H = U_1 \times (t - t_o) = 0.12 \times 70 = 8.4 \text{ Btu}$$
$$H = f_i \times (t - t_1) = 1.34\ (70 - t_1) = 8.4$$
$$1.34\, t_1 = 93.80 - 8.4 = 85.40$$
$$t_1 = 63.7 \text{ F inside wall surface.}$$
$$H = f_o \times (t_2 - t_o) = 4.02\ (t_2 - 0) = 8.4$$
$$4.02\, t_2 = 8.4$$
$$t_2 = 2.09 \text{ F outside wall surface}$$

Wall No. 2

$$H = U_2 \times (t - t_o) = 0.24 \times 70 = 16.8 \text{ Btu}$$
$$H = f_i \times (t - t_1) = 1.34\ (70 - t_1) = 16.8$$
$$1.34\, t_1 = 93.8 - 16.8 = 77.0$$
$$t_1 = 57.4 \text{ F inside wall surface.}$$

The outside t_2 may be calculated as before, or since

$$(t_2 - t_o) = \frac{1}{3} \times (t - t_1) = \frac{1}{3} \times 12.6 = 4.2 \text{ F.}$$
$$t_2 = 4.2 \text{ F outside wall surface.}$$

Note:—Actual surface temperatures read directly from the curves of Fig. 5 for a 70 F indoor-outdoor difference and corrected to a breathing line air temperature of 70 F and outside air temperature of 0 F for a wall similar to No. 2 show:

t_1 (measured) $= 60 - 2.5 = 57.5$ F inside surface (calculated $= 57.4$ F).

t_2 (measured) $= 9 - 2.5 = 6.5$ F outside surface for an average wind velocity of about 10 mph. At a wind velocity of 15 mph. the outside surface temperature would have been decidedly lower, and would have more nearly agreed with the calculated value of 4.2 F. It is to be noted that the inside surface temperature would have been lower also at a wind velocity of 15 mph. but it would only be affected about one-third as much as the outside surface temperature, and would still be in good agreement with the calculated value of 57.4 F.

It will be evident that with a portable thermocouple outfit for actually measuring t_1 and t_2 on a zero day with the breathing line temperature of the air in

[6] Data for calculation taken from Chapter I of THE GUIDE, 1929, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.

house at 70 F, one could readily rate the *actual* wall *as built* against walls No. 1 and No. 2. The air temperatures selected do not have to be 70 F and zero, but are whatever exists on the day of the test, although a fairly low outside temperature is desirable, and the surface temperatures should be taken on a north wall on a cloudy day or at night.

No. 871

HOW COMFORT IS AFFECTED BY SURFACE TEMPERATURES AND INSULATION

By PAUL D. CLOSE,[1] NEW YORK

MEMBER

The importance of a hitherto neglected factor in the design of a heating system and the maintenance of the proper atmospheric conditions in a building, has recently been emphasized by Professors Willard and Kratz of the University of Illinois.[2] This factor relates to the radiation of heat from the body to cold surfaces, producing a feeling of chilliness, even under conditions which ordinarily would be considered satisfactory from the comfort standpoint. This paper contains a discussion of bodily comfort, including the effect of the heat emitted by radiation, as well as a mathematical analysis of surface temperatures, the direct relation of insulation thereto, and the consequent indirect relation of insulation to comfort.

UNTIL a few years ago it was thought that the comfort of our atmospheric environment depended solely on the temperature registered by the dry-bulb thermometer. However, the investigations at the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS,[3] and at the *Harvard School of Public Health*, have disclosed the fact that the sensation of comfort depends not only on the dry-bulb temperature of the air, but also on the amount of moisture it contains and the rate of air motion. Dry air at a relatively high temperature may feel cooler than air of considerably lower temperature with a high moisture content, and air motion makes any moderate condition feel cooler. These tests were used as the basis of an experimentally determined scale, known as *effective temperature*, which is a true measure or index of a person's feeling of warmth in all combinations of temperature, humidity and air motion.

RELATION OF WALL AND CEILING TEMPERATURES TO COMFORT

As pointed out by Professors Willard and Kratz, there is also another factor to be considered in many cases, namely, the temperature of surrounding objects, particularly the wall, window and ceiling surfaces, if adjoining unheated spaces. In the process of metabolism, heat is continually being radiated from the body to the colder surrounding surfaces. Since, according to the Stefan-Boltzman law, the rate at which heat is emitted from the body by radiation is

[1] Technical Secretary, AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS.
[2] See *Wall Surface Temperatures* by Willard and Kratz, p. 447.
[3] See A. S. H. V. E. TRANSACTIONS, Vols. 28 to 34 inclusive.

Presented at the Semi-Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS, Minneapolis, Minn., June, 1930.

proportional to the difference of the fourth powers of the absolute temperatures, the effective temperature scale applies only when the temperature of the surrounding objects is at or near the surface temperature of the skin, and does not take into consideration any appreciable difference which may exist between the wall and skin temperatures.

Valuable information on the subject of body radiation is contained in *Smithsonian Institution* Publication, No. 2980, by L. B. Aldrich, entitled, A Study of Body Radiation. Reference is made to experiments conducted in the Nutrition Laboratory of *Carnegie Institution* in Boston, and in the *Smithsonian Institution* in Washington. The Smithsonian experiments were made at the

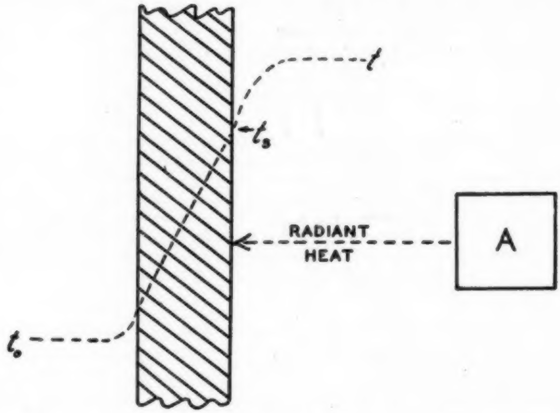

FIG. 1. SECTION THROUGH WALL SHOWING TEMPERATURE GRADIENT

request of the *New York Commission on Ventilation,* and the results of these experiments, as well as those conducted at the *Carnegie Institution,* are summarized in the aforementioned publication as follows:

1. The radiation from the skin and clothing is approximately that of a *black body* or perfect radiator.

2. A cloth-covered, vertical, cylindrical calorimeter at body temperature loses in still air 60 per cent by radiation and 40 per cent by convection. A similar horizontal calorimeter loses 54 per cent by radiation and 46 per cent by convection. The human body convection loss is probably similar to this, that is, the convection loss is roughly one third less than the radiation loss, in still air and normal room temperatures.

3. Increasing air motion rapidly decreases the percentage radiation loss and increases the convectional.

4. Total body radiation similarly decreases with air motion.

5. Increase in room temperature (which also means increase in wall temperature) produces a progressive lowering of radiation loss.

6. Keeping room and wall temperatures unchanged, the temperature of skin and clothing decreases with increasing air motion, the decrease being greatest on the side facing the wind and about one half as great on the side away from the wind. The clothing temperature drop on the side towards the wind is about one third greater than the corresponding skin temperature drop.

7. At normal indoor temperature, in still air and with the subject normally clothed and at rest, body heat losses are distributed as follows:

Evaporation of water	24 per cent
Radiation	46 per cent
Convection	30 per cent

8. The air temperature falls to room temperature very rapidly as the distance from the body increases. That is, there is a steep temperature gradient in the first ½ in. or so from the body surface. With the thermoelement about 12 in. away, no effect of the presence of the body could be detected.

9. The radiation loss from a nude subject is about twice as great for a room temperature of 59 F as it is for a room temperature of 79 F.

10. Normal fluctuations in humidity indoors produce negligible effect upon the radiation loss. Were the air of the room exceedingly dry, changes might be noticeable.

Particular attention is called to Item 5 to the effect that an increase in wall temperature produces a decrease in the radiation loss. Consequently, if the wall and ceiling temperatures are increased sufficiently, a lower room temperature is permissible, provided the relative amount of window surface is not excessive. This is one of the principal features of the panel system of heating extensively in use in England,[4] in which the heating pipes are embedded in the walls or ceiling, and hot water forced through them to maintain the surfaces at a temperature of about 75 F. The increased surface temperature decreases the heat loss from the body by radiation, and thereby permits the maintenance of a lower room temperature, which results in fuel economy.

Attention is also called to Item 9 to the effect that the radiation loss was about twice as great for a room temperature of 59 F as it was for a room temperature of 79 F. This result was obtained from experiments conducted by Dr. C. G. Abbott and Dr. F. G. Benedict, in the *Carnegie Institution* in Boston. That the radiation loss increased with the decrease in the room temperature was no doubt due primarily to the fact that the lower room temperature resulted in a lower wall temperature, with a consequent increase in the difference between the wall, window and skin temperatures. Of course, lowering the room temperature would also tend to lower the skin surface temperature, but on a cold day the wall surface temperature apparently decreases to a greater extent than the skin surface temperature for a given reduction in the inside air temperature. According to Du Bois, when the room temperature is lowered, one changes one's integument into a *suit of clothes* and restores the zone where the blood is cooled from the skin to a level some difference below the surface.

If the temperature of the interior surfaces is sufficiently lower than the body surface temperature, the feeling of chilliness will result even though the normal effective temperature is maintained. This, as previously stated, is due to the

[4] See Panel Warming, by L. J. Fowler, A. S. H. V. E. Journal (*Heating, Piping and Air Conditioning*), January, 1930.

increase in the loss of heat from the body by radiation. Since the rate at which heat is given off from the body by radiation increases as the temperature difference between the body surface and the wall and ceiling surfaces increases, the sensation of chilliness will increase to some extent as the wall surface temperature decreases, unless the increased loss of bodily heat by radiation

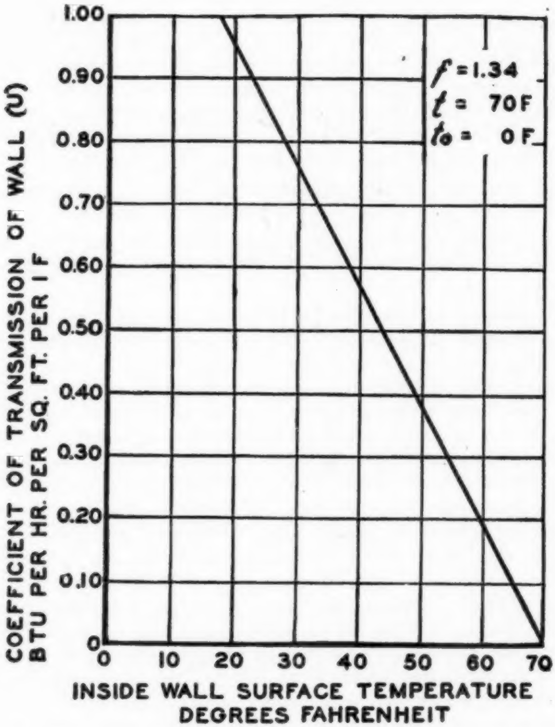

FIG. 2. CURVE SHOWING RELATION BETWEEN INTERIOR SURFACE TEMPERATURE OF WALL AND COEFFICIENT OF TRANSMISSION OF WALL WHEN f, t AND t_o ARE CONSTANT AND HAVE THE VALUES INDICATED

is counterbalanced by an increase in the room temperature. Likewise, the less the heat resistance of the wall, the lower the interior wall surface temperature for the same outside temperature, and the greater will be the sensation of chilliness for the same effective temperature. Consquently, if the walls of a building are poorly insulated, or if they contain a large percentage of window surface, the effective temperature must be increased in cold weather to offset the increased loss of heat from the body.

EFFECT OF INSULATION ON SURFACE TEMPERATURE

Inasmuch as the wall surface temperature is a function of the overall heat resistance of the wall, the addition of a given thickness of insulation will materially increase the surface temperature, and permit the maintenance of the normal effective temperature, provided the relative amount of window surface is small. Under the circumstances, the insulation will result in a double heat saving, *first,* that due to the lower temperature difference, and *second,* that due to the retarding of the rate of heat transfer through the wall.

A study of the variation in the wall surface temperature with the various

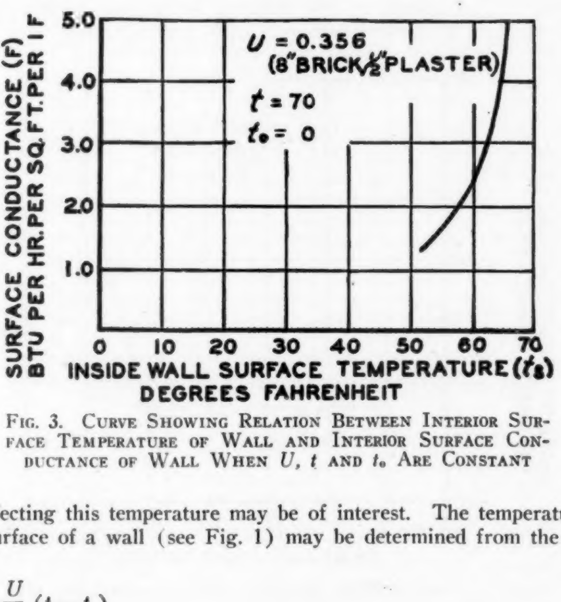

FIG. 3. CURVE SHOWING RELATION BETWEEN INTERIOR SURFACE TEMPERATURE OF WALL AND INTERIOR SURFACE CONDUCTANCE OF WALL WHEN U, t AND t_o ARE CONSTANT

factors affecting this temperature may be of interest. The temperature of the interior surface of a wall (see Fig. 1) may be determined from the following formula

$$t_s = t - \frac{U}{f}(t - t_o) \qquad (1)$$

where—

t_s = temperature in degrees Fahrenheit of the interior wall surface

t = inside air temperature

t_o = outside air temperature

U = coefficient of transmission of wall in Btu per hour per square foot per degree Fahrenheit difference in temperature

f = film conductance of interior wall surface expressed in Btu per hour per square foot per degree Fahrenheit difference in temperature between the air in the room and the wall surface.

From formula (1) it is apparent that the wall surface temperature is a function of the inside and outside air temperatures, the coefficient of heat transmission of the wall and the interior surface or film conductance of the wall. Sun effect also has a bearing on the inside surface temperature, but is neglected

in this analysis. Consider an 8-in. brick wall with ½ in. of plaster applied directly to the interior surface, the coefficient of transmission of which (according to the A.S.H.V.E. GUIDE 1930) is 0.356 Btu per hour per square foot per degree Fahrenheit difference in temperature. If the inside and outside temperatures are 70 F and 0 F, respectively, and the surface conductance is 1.34, the temperature of the interior wall surface will be

$$t_s = 70 - \frac{0.356}{1.34}(70 - 0) = 51.4\text{ F}$$

VARIATION IN SURFACE TEMPERATURE WITH VARIOUS FACTORS

The curve (Fig. 2) illustrates the variation in the wall surface temperature with the coefficient of transmission (U) of a wall, assuming the inside and

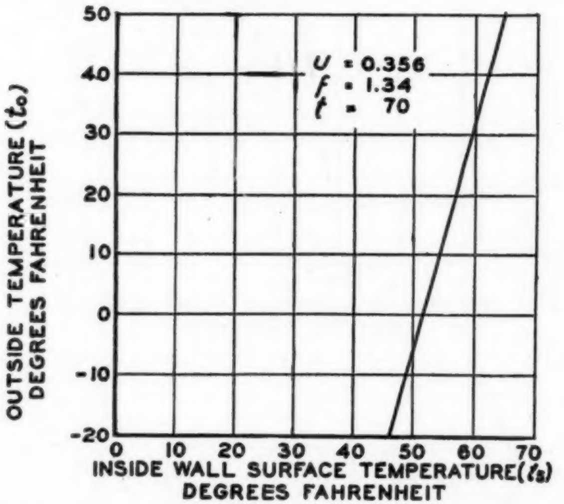

FIG. 4. CURVE SHOWING RELATION BETWEEN INTERIOR SURFACE TEMPERATURE OF WALL AND OUTSIDE TEMPERATURE WHEN U, f AND t ARE CONSTANT AND HAVE THE VALUES INDICATED

outside air temperatures to be constant at 70 and 0, respectively, and the surface conductance to be 1.34 (average value used in The A.S.H.V.E. GUIDE 1930). It will be noted that if U has a value of 1.00—which is the approximate coefficient of a sheet metal wall (flat surface) when exposed to a wind velocity of 15 miles per hour—that the wall surface temperature is only 17.8 F. If the curve were extended, t_s would be equal to t_o when $U = f$ or 1.34. As the coefficient of the wall (U) approaches 0, the inside surface temperature approaches the inside air temperature (t) which in the case is 70 F. If the 8-in. brick wall under consideration is insulated with 1-in. of rigid insulation having a conductivity of 0.33, so that the coefficient is reduced to 0.148, the inside surface temperature would be 62.3 F for the conditions

involved. Hence, the insulation in this case, together with the air space resulting from the installation thereof, would increase the surface temperature 10.9 F on a zero day, and would therefore have a marked effect upon the degree of comfort of the room, because of the resulting reduction in the loss of heat by radiation. An increase in the surface conductance increases the surface temperature when the other factors remain constant, as will be seen from Fig. 3. For a still air conductance of 1.34, an 8-in. brick wall with ½-in. of plaster, will have a surface temperature of 51.4 F as previously stated. If the velocity of air passing over the surface is increased, the surface conductance will increase. At a conductance of 2.00, which would result from a slight wind movement, the surface temperature will be increased 6.1 degrees to 57.5 F, assuming the inside and outside temperatures to be 70 F and 0 F, respectively. Although the value of U will increase slightly with an increase

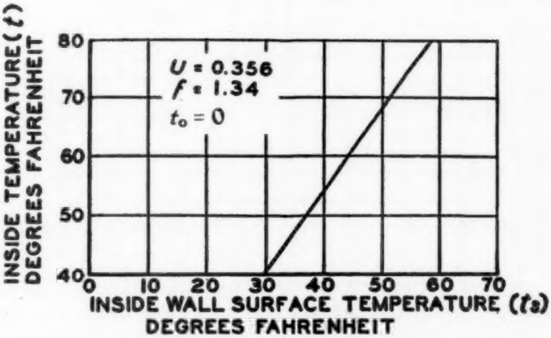

FIG. 5. CURVE SHOWING RELATION BETWEEN INTERIOR SURFACE TEMPERATURE OF WALL AND INSIDE TEMPERATURE WHEN U, f AND t_o ARE CONSTANT AND HAVE THE VALUES INDICATED

in the surface conductance, it is assumed constant in this case, as this assumption will not appreciably alter the results. If the velocity of the air passing over the interior surface is increased to about 15 mph, the surface conductance will be 4.02, assuming the 3 to 1 ratio used in THE GUIDE 1930, and the surface temperature will be about 63.8 F, an increase of 12.4 over the still air condition.

It is apparent that an increase in the outside temperature will increase the interior surface temperature, whereas a decrease in the inside temperature will decrease the interior surface temperature. These relationships are shown graphically by Figs. 4 and 5. For the type of wall under consideration (8-in. brick, ½-in. plaster), the inside surface temperature increases from 46 F to 64.7 F when the outside temperature is 50 F, assuming that the inside temperature remains constant at 70 F. If the inside air temperature varies, the wall surface temperature will increase from 29.4 F when the inside temperature is 40 F, to 58.8 F when the inside temperature is 80 F, assuming that the outside temperature remains constant at 0 F.

The inside temperature is usually about 70 F and the surface conductance may be considered constant for walls and ceilings, and having a value of 1.34. Hence, in most cases the only variables to be taken into consideration are the outside temperature and the coefficient of transmission of the wall. The chart

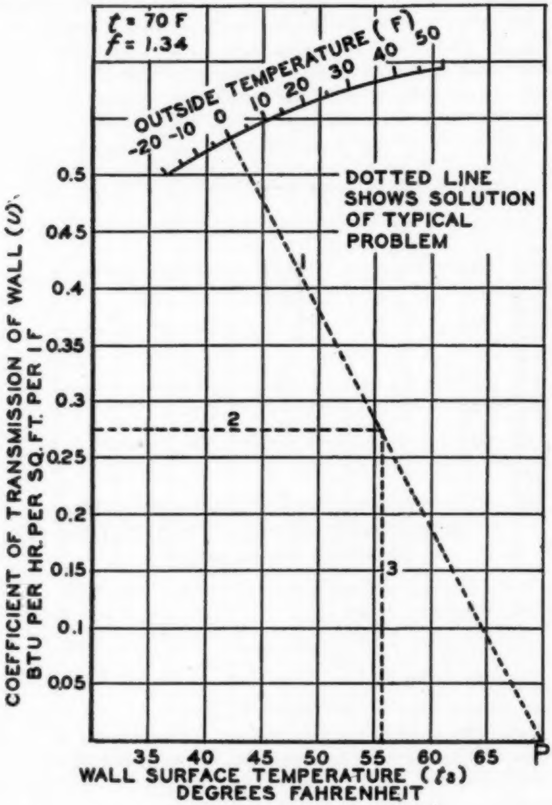

FIG. 6. CHART FOR DETERMINING WALL SURFACE TEMPERATURE FOR VARIOUS OUTSIDE TEMPERATURES AND WALL COEFFICIENTS

(Fig. 6) may therefore be used for estimating the wall surface temperature for any combination of these two variables.

WINDOWS

Window surface temperatures for single glass are considerably lower than wall surface temperatures on a cold day. Hence, if the exterior walls of a room contain an excessive amount of glass, double or triple windows will materially reduce the loss of heat from the body by radiation. The coefficient

of transmission (U) of single glass, based on a wind exposure of 15 mph, is 1.13 (A.S.H.V.E. Guide 1930) and the still air surface conductance (f) is 1.50. Hence, on a zero day, with an inside temperature of 70 F, the glass surface temperature will be—

$$t_s = 70 - \frac{1.13}{1.50}(70 - 0) = 17.3 \text{ F}$$

With double glass, the surface temperature will be 49 F, based on a coefficient of transmission of 0.45, and with triple glass the surface temperature will be 56.9 F, based on a coefficient of 0.281. Therefore, double glass used in place of single glass will increase the surface temperature on a zero day to the extent of 31.7 deg, whereas triple glass will increase the surface temperature 39.6 deg. Consequently, the effect of double or triple glass in the case of windows is even more pronounced than the effect of insulation in the case of walls; hence the importance of using double or triple glass in a room having a large amount of window surface.

Insulation and Heat Loss by Radiation

The Stephan-Boltzman radiation law may readily be applied to a typical problem involving the change in wall surface temperature resulting from the installation of a given thickness of insulation. The metabolism of the human body is somewhat complicated. Heat is given off not only by radiation but also by convection and evaporation of water. The physiological processes are continually striving to adjust themselves to the surrounding atmospheric environment in order to equalize the heat produced and the heat given off from the body, and thereby to maintain bodily comfort. Hence, any change in the loss of heat by radiation may also involve a change in the loss by convection and evaporation as well as a change in the temperature of the exposed skin. These changes do not readily lend themselves to mathematical analysis, and for the sake of simplifying the calculations, an inanimate object instead of the human body will be considered in the following problem.

Assume the surface of an object (A, Fig. 1) to be maintained by electrical means or otherwise at a constant temperature of 90 F in a room temperature of 70 F. Except for the changes in physiological reactions, this object may be considered analogous to a human body in this analysis. The problem is to determine the decrease in the rate of heat emission by radiation from this object to the exterior walls on a zero day resulting from the installation of a 1-in. thickness of a flexible insulation to the brick veneer frame walls of the room.

The coefficient of transmission of the uninsulated wall is 0.247 (A.S.H.V.E. Guide 1930), and the surface temperature, according to formula (1) is approximately 57 F. The absolute temperature, therefore, is 57 + 460 or 517 deg. The coefficient of transmission of the insulated wall is 0.116, the surface temperature, 64 F, and the absolute temperature, 524 deg. The decrease in the loss of heat by radiation, according to the Stephan-Boltzman law:

$$= \frac{[(5.50)^4 - (5.17)^4] - [(5.50)^4 - (5.24)^4]}{(5.50)^4 - (5.17)^4}$$

$$= \frac{200 - 160}{200} = 0.20 \text{ or } 20 \text{ per cent}$$

Therefore, under these conditions, the insulation would reduce the rate at which radiant heat is absorbed by the wall to the extent of 20 per cent. While

this analysis would not apply exactly to a human being, the result is indicative in a measure of the effect of the insulation on comfort. An increase in the loss of heat from the body by radiation would tend to lower the skin temperature. If the temperature of the object were only 75 F, the insulation would reduce the rate of heat loss to the wall by radiation, 39.2 per cent.

Consider the problem from another angle. To what degree would it be necessary to raise the inside air temperature in the case of the uninsulated wall (disregarding windows) in order to obtain the same rate of heat emission by radiation as in the case of the insulated wall?

In solving this problem, it is merely necessary to equate the radiation emission (Q) in the two cases, and solve for t as follows:

$$Q = D\left[\left(\frac{460+90}{100}\right)^4 - \left(\frac{460+t_s}{100}\right)^4\right]$$

$$= D\left[\left(\frac{460+90}{100}\right)^4 - \left(\frac{460+64}{100}\right)^4\right]$$

where—

D = radiation constant

t_s = wall surface temperature

= 64 F

From formula (1)—

$$t_s = t - \frac{0.247}{1.34}(t-0)$$

$$= 0.816\ t \text{ and}$$

$$t = 78.5 \text{ F}$$

It would therefore be necessary to increase the room temperature (t) from 70 F, in the case of the insulated wall, to 78.5 F, in the case of the uninsulated wall, to obtain the same heat loss from A (Fig. 1) by radiation. Of course, it probably would not be necessary to increase the air temperature to this extent to produce the same bodily comfort—in fact, this dry bulb temperature without a correspondingly low wet-bulb temperature would ordinarily produce a feeling of lassitude or discomfort—but the result gives a hint of the importance of insulation from the standpoint of winter comfort, and the necessity of increasing the temperature in the zone of occupancy of an uninsulated room to offset the increase in the loss of heat from the body by radiation.

JOINT DISCUSSION

Wall Surface Temperatures

and

How Comfort Is Affected by Surface Temperatures and Insulation

J. D. HOFFMAN (WRITTEN): Professors Willard and Kratz have here offered a simple adaptation of the temperature-drop curves to the rating of building walls. Trying to convince a person of the probable heat losses through the walls of his own home by making a few calculations on a piece of paper is, in most cases, a waste of valuable time. On the other hand, a simple, practical demonstration as indicated in the last paragraph of this paper would be an object lesson that would carry conviction to the most skeptical. Wall and floor insulation for residences is being accepted these days as a desirable thing, but the big question with the house owner is, what kind of insulation? With the plan suggested it should be an easy matter to compare one wall type wth another under any given set of weather conditions and draw reasonably accurate conclusions as to their relative merits.

C. H. B. HOTCHKISS (WRITTEN): This paper contains the first data resulting from field studies which may be readily checked with a similar laboratory set-up whereby the effects of the several variables may be separated and analyzed individually.

Under item 3 of the Summary of Results, the authors state: "At any outside air temperature, the inner air to surface difference in temperature is greater than the outer surface to air difference in temperature, except for still air conditions.

"An inspection of Fig. 3 at any indoor-outdoor air temperature difference shows the temperature difference *A* is always greater than *B*."

Fig. 5 however does not bear this out for at 11:00 a. m. on January 7, 1929 the *outer surface to air* difference is considerably greater than is the *inner air to surface* difference. Consequently it would seem that this should be taken into account in the statement.

Under item 7 of the Summary of Results the authors call attention to the following: "It is noteworthy, however, that the outside surface temperature of the south wall was always slightly higher than the outside surface temperature of the north wall even with no sunshine and at night."

Fig. 5 shows that during the period when these data were collected there was a wind movement of from 3 to 18 mph and that while this wind was from the south for some of the time, there was no north wind. Consequently the higher outside surface temperature of the south wall occurred *in spite of the effect of this wind.* If the reasoning applied elsewhere in the paper and as brought out by Fig. 4 is valid, then if both the north and south walls had been in still air the outer surface temperatures of the south wall should have been not slightly, but considerably, higher than those on the north wall. Would this not tend to show that there may be *considerable* difference in outer surface temperature between the north and south walls on cloudy days *when there is little wind movement?* Also may it not be taken to mean that the effect of sunshine may

extend for considerably longer periods than the example shown would indicate? Also if looked at in this light it should tend to strengthen conclusion 4.

In the numerical examples worked out near the end of the paper the authors calculate the inside and outside surface temperatures of a wall having a value of $U=0.24$ Btu. They do this by the use of data which assume *a condition of steady flow*. They then check these calculations by data taken from Fig. 5, which were collected during a period when the outside air temperature was changing and when it is reasonable to assume that a condition of steady flow through the wall did not exist. The agreement must be accounted for either on the grounds that the outside air temperature was sensibly constant at the time of observation or else that the element of time lag in this particular wall is slight and that steady flow is quickly set up in it. One cannot help wondering if a field observation similar to that used on the wall of $U=0.24$ would yield a similarly close agreement if applied to a wall with $U=0.12$ which could be obtained only by adding greater bulk of the same material to the existing wall or by substituting materials of lower conductivity. Would not the matter of time lag and thermal storage make it difficult to apply data obtained under steady flow conditions to such a wall when operating under the influence of varying outside weather conditions? It is apparent that the use of a field outfit could be readily applied to walls of frame residences. Until further information is at hand though it seems doubtful if such a method should be applied to walls of other types.

It also seems that the use of a portable outfit might lead to discrepancies unless a procedure can be worked out to insure the observation being continued over a long enough period to nullify the possibilities of accidental readings such as might result if the apparatus were set up for a short time only. The necessity for this is quite clearly shown by Fig. 3. In this figure it will be noted that there is considerable variation in the readings of the temperatures obtained at practically like temperature conditions. The plotted points do not follow the mean lines but diverge from them in both directions. Readings taken over a short period might thus prove misleading if they should strike a certain set of momentary conditions. It would seem that the determination of a suitable procedure should not be a difficult matter, and the obvious advantages of such a method of rating walls ought to make it well worth any additional attention necessary to make it fully workable.

JOHN HOWATT (WRITTEN): The effect of cold walls and cold furniture upon the occupants of a room has not always been recognized nor considered of the importance it actually is. All objects in a room radiate heat to each other. In the interchange the warmer objects lose heat to the colder. The occupants of a room, therefore, give off heat to the colder walls and furniture, the rate being dependent upon the differences in their temperatures. In intermittent heating such as is practiced in churches and school buildings it is found necessary to carry the room temperature at the desired standard for some time before occupancy to warm furniture and walls, otherwise the occupants of the room will feel uncomfortably cold even though the air temperature as indicated by a dry-bulb thermometer is 70 F. The length of time required to warm the furniture and walls depends upon the materials and their volume. It is a rule in our school work to require that room air temperatures be maintained for 30 min before opening of classes in the morning.

All studies of recent years emphasize the necessity of good outside wall construction. In order that maximum comfort shall be obtained the difference between the inside air temperature and the inside wall temperature should be small. The experiments of Professors Willard and Kratz show that this difference is greatest when the wall insulation is poorest, and least when the overall thermal effectiveness of the wall is best. Economy and comfort both demand an outside wall construction efficient in preventing heat transfer. This informative paper points the way to measure such efficiency.

The temperature gradients (Fig. 4) show a very abrupt rise from the inside wall surface to the room air temperature. One would expect the cold wall surface to affect the air temperature to a greater distance than the curves indicate. The factor of infiltration, always a very doubtful one, will materially change the distance from the wall where the air will be at room temperature. Figs. 4 and 6 show that wind increases the inside-surface-to-air difference which is to be expected as the rate of infiltration increases at the same time. Tight wall construction will help. The work of scientists and designers may be brought to naught by the indifferent workmanship of an incompetent or unscrupulous builder. Better building construction is the need of the industry today. The bricklayer on the job is not concerned with heat transfer coefficients so much as getting the wall up. It is essential of course that all coefficients of heat transfer be known, as without complete understanding little progress can be made. But after all wall coefficients have been calculated, a generous allowance must be made to cover the frailities of the builder because of the human element that must be considered on every construction job. A factor of safety must be added to take care of the incalculable contingencies that experience has taught us will have to be met.

F. B. ROWLEY (WRITTEN): This paper is a contribution of exceptional value to the literature on heat transmission through buildings. It brings out the significance of inside surface temperatures, their relation to comfort and the manner in which they are affected by wall construction and outside weather conditions. Everybody is familiar with the fact that comfort in cold weather requires warmer air temperature than in mild weather. This has generally been attributed to lower relative humidities. It is now pointed out that another reason for this condition is the lower temperature of the surrounding wall surfaces. For a given wall, the inside surface temperature will drop with the outside air temperature and wind velocity. The greater this drop, the greater the temperature rise of the inside air must be to counteract radiation effects.

The amount of the inside surface drop will depend not only upon the outside weather conditions but also upon the construction of the wall, its resistance to heat transmission and air infiltration. If these resistances are high, the inside surface temperature will be more nearly equal to the inside air temperature and the final effect would be a lower required air temperature in the room.

Referring to Fig. 4, it is significant that the wind velocity on the outside surface has lowered the inside surface temperature approximately the same number of degrees as it has the outside surface temperature. This seems to be explained by air leakage through the outside surface of the wall. If there had been no such leakage we should expect the inside surface temperature drop to be much less than the outside surface temperature drop. The effect of this apparent air leakage may be shown by calculating the percentage increase in

heat transmission from the increase in temperature drop through the various sections of the wall. Since the only change of conditions affecting the transmission of heat through the wall in the two cases is the outside wind velocity, it may be assumed that all other coefficients except the outside surface coefficient remain substantially the same and that the heat flow through the wall in either case is proportional to the temperature drop through the various sections. This being the case the increase in heat flow as indicated by the difference in temperatures between the air on the inside of the room and the outside surface temperature equals:

$$1 - \frac{72.5 - 8}{72.5 - 14} = 10 \text{ per cent}$$

The increase in heat flow indicated by the drop in temperature from the air to the inside wall surface temperature equals:

$$1 - \frac{72.5 - 55}{72.5 - 60.9} = 41 \text{ per cent}$$

This difference can be explained by the air leakage through the outside surface of the wall and it indicates the necessity of a good wind stop in the outside surface construction. If the wind leakage through the complete wall was determined it would undoubtedly be very low due to the inside surface of lath and plaster. However, the damage is done when the air leaks through the outside surface and cools the air space between the studs.

Referring to Figs. 3 and 6, if the various curves are extended to the left they intersect each other substantially at the zero point. With the exception of the top curve on Fig. 3, they are all straight lines and there is a constant ratio between the coefficients for the different portions of the wall at different mean temperatures. As pointed out by the authors this constant ratio does not appear to hold for the inside and outside surface coefficients where the wall is exposed to sunshine as was the case for the wall of Fig. 3. This is the one exception to the straight line relations between the various coefficients and I believe should have additional study.

W. H. CARRIER: I would like to commend the authors of this paper on the manner of presentation. The answer should come first and the reasons for it afterward, as they have done. I would recommend to the authors that an addenda accompany the paper describing more minutely the equipment used, also a discussion of the accuracy and the reason for the adaptation of the equipment.

Man was not born with clothing, with shelter or with fire. Presumably when he was born he was relatively naked except for the good thatch on his head to protect him from the burning sun and perhaps a little more natural clothing on his body than we have today through our process of evolution. He was born for the conditions that surrounded him.

Horse and man are the only two animals that have their principal means of getting rid of heat under emergencies by means of perspiration. They thrive in hot weather, although they do not like it. The horse lathers in hot weather; a man perspires freely. You do not find that among any other animals. Our nearest relatives (I think they are a long ways away) are a jungle animal; they will die out in the sun; they are not our ancestors. We were generated in a different climate. We were driven out of the Garden of Eden, so to speak, by necessity, perhaps a change of climate, and therefore we have developed our

wonderful intelligence and we have been adapting ourselves to environment ever since.

Shelter and clothing perhaps came first. The matter of producing fire was one of the great inventions of man. No other animal has it and our Society is founded on the development of the use of fire. Until recently fire was the only thing we wanted; now we acknowledge the part that humidity plays as well as temperature.

We are getting more complex all the time. We have to know how hot the wall is and how cold the floor is, how hot the ceiling is and what the temperature of the air is. If we are going to keep all these things in mind we will need an engineer for every room of the house. We will need thermocouples everywhere to be sure that we will be comfortable, because there is no uniform temperature in any part of the room-walls, ceiling or anything else—and we have to evaluate glass factors, etc. How are we going to do all this?

Well, fortunately, I believe it is a very simple thing, in spite of the difficulties that have been pointed out in these various papers and discussions. The exactness of the comfort chart has been challenged. It is suggested that temperature has one effect and radiation from the walls and windows another effect. Nothing has been said, however, about the lower temperature which frequently exists at the floor and which is a larger area than the outside walls. Floor temperatures are sometimes pretty cold due to cold air seeping in. Take a room like this—the temperature of the floor will be far more important from the standpoint of radiation than the outer walls. It is not a question of radiation. I get radiation from my head to the floor and my head is on top. My feet as well as my head are warmed by the ceiling because of the angle of radiation. It would take a mathematician to find out whether you are going to be comfortable or not and by the time you got it all figured out, well, it would be summer! It is a very complex thing. You do not want to forget that while we have cold walls, cold windows, and cold floors, you frequently get hot ceilings and the hot ceiling has just as much countereffect as a cold wall and perhaps more so in the radiation effect. You might even have cold walls and windows but due to the hot ceiling you would be too warm.

Nobody said anything about ceiling. We are not anywhere near the ceiling. The air near the ceiling does not affect us, but the radiation does, and the difference between the ceiling temperature and body is usually as much as the walls, neglecting the windows. What is the temperature of a room? We have all assumed that we have been talking about the temperature of a room. Nobody has defined what the temperature of a room is. Had we not better define what we are talking about? The temperature of the room is not the temperature of the air. It is not the temperature of the walls. It is not the temperature of the ceiling. It is the temperature that is affected by all of us.

If we use an extremely minute resistance wire, or a thermocouple of similar dimensions, we will get nothing but the temperature of the air itself. If we use the ordinary thermometer or thermostat or an object the size of the human body, we get the temperature of radiation and of the air; and we get radiation from all parts, not from any one part. Now if we use something in between, perhaps we get something nearer the air temperature instead of this composite temperature. What are we going to use as a standard of temperature when there are no two thermometers, unless they are identical, that will give us exactly the same

results? At low temperatures thermometers probably measure fairly accurately the lower radiation effect, the low waves. Glass absorbs low waves. So, whether we blacken the thermometers or not, I do not believe that the temperature at which we feel comfortable makes much difference. If you had a fire, however, over here and you hold a thermometer there, there would probably be quite a difference, whether the temperature was measured with a blackened thermometer or a lightened one, but most of our radiation (and the same applies to a radiator) is due to a small difference in temperature. Low wave lengths are absorbed largely by the glass. The absorption of glass under those conditions is large and therefore the ordinary thermometer and the thermostat of dull metal, is as much subject to the effects of radiation as it is to air temperature, just like the body in that respect.

I will agree that you will be very uncomfortable if you sit next to a fireplace with the heat pouring at you on one side and no heat in the room and you have a cold window or a wall at your back.

The thermometer might read the same; you might have an overall transmission the same but the transmission at your back is so high that you are uncomfortable; you are too hot on one side and too cold on the other. This is one reason why we should not have any great differences of temperature between walls, floors and air. But the thermometer does show what the average condition is; it integrates all effects, just exactly as the body does. The fact that we are giving out some heat does not enter into the problem. The fact that our temperatures are different does not enter into the problem. It is the differential between an unheated body and ourselves that controls our radiation and it is the difference between the room air temperature and our body that controls our convection.

I have made some intensive studies on the effect of radiation errors on the wet-bulb thermometer and I believe the integration effect of a thermometer due to convection can be taken as an indication of what the body is affected by. You cannot put it on the wall or you cannot put it over a window. You cannot put it up to the ceiling. You have got to put it where you are. A wet and dry-bulb thermometer placed where you are will show the effective temperature you are subjected to, whether you have a fireplace on one side and a big window expanse on the other, or are surrounded by walls and air of uniform temperature. A wet and dry-bulb thermometer reacts to radiation effects and integrates the equivalent result.

A wet-and dry-bulb thermometer under any variable condition will, I believe, give you exactly the result obtained in the Laboratory, effective temperature tests where the walls, ceiling and air were purposely kept all at the same temperature so as not to complicate the problem.

If you take your temperature in another way, it is not correct. If you use exceedingly fine resistance wire, then you measure the air temperature and you do not measure integrated effects. That is why I say we have to have an addendum to this paper to know the details. What gage wire was used in these thermocouples? It makes a difference.

Everybody here has talked about temperature of a room and they have assumed that is the temperature of the air. It is not any such thing! It is a composite effect. The difference in convection between a thermometer and the human

body is not very great. The wet-bulb thermometer responds just the same as tne wetted surface of your skin.

So I think we should be careful not to, when you say *temperature of a room,* mean the temperature over at the wall, where you may have the thermometer. It certainly is not the temperature of that wall or the thermometer on that wall. It is the temperature where you are. Nor is it the temperature at the window. Nor is it the temperature of the ceiling or the temperature of the floor, and we do not have to calculate and integrate all these values when a thermometer does it for us.

If my assumptions of this integration and effects of thermometers is correct, both wet and dry-bulb, then I have answered the uestions that have been raised concerning effective temperatures. For greatest comfort you have to have a fairly uniform temperature all over. You do not want a hot head and cold feet, nor do you want a cold back and overheated on the other side, although you might have exactly the same effective temperature and the heat emission will be the same. Uneven distribution of temperature is exceedingly uncomfortable.

It is not a question of the removal of heat; it is a question of the removal of heat on one side, and therefore, all that has been said on good insulation holds, and double windows too, if you are going to sit near them, and perhaps counteracted by a certain amount of radiation but not in the way it was presented.

F. C. HOUGHTEN: These two papers give us additional data on two major subjects in which the Laboratory and the Society have been interested, namely, heat transmission and comfort. The data on relation of wall surface temperatures and air temperatures, both inside and out, to the rate of heat transfer and the ratio of the ouside difference to the inside difference are valuable in our consideration of heat transmission. No doubt considerable use will be made of these data in evaluating certain walls installed, as pointed out by the authors.

I was interested in the ratios of the outside temperature difference, between wall and air to the inside temperature difference. This ratio is given as about three or the same value as has been accepted in the past for the ratio of the outside surface transfer coefficients for a 15-mile wind, to the inside coefficients for still air. In general it is assumed that this difference is due to difference in air velocity and radiation.

There is another factor that may account for the ratio of three and I would like to hear further from the authors concerning their consideration of this factor. Heat capacity of a wall is becoming recognized as a very important factor in heat transfer in the winter when there is a temperature drop outward throughout the day, and more important perhaps in the summer time when there is a cyclic change in the direction of the temperature drop, due to sun effect and diurnal difference in temperature. It has been found from the work of the Laboratory that as the temperature changes throughout the day, the relation between the heat flow through the outside surface and the inside surface changes. If the factor of three were determined in the winter time, during the falling temperature part of the cycle, there would be a greater heat transfer, through the outer surface than through the inner surface, which might account for part of that factor of three.

Considering the other important point brought up in this paper, the effect on comfort: Mr. Carrier and his committee, working with the Laboratory in pre-

paring plans for the investigation of comfort some ten years ago, and the Laboratory in carrying out this program, and later in publishing the findings, did give considerable attention to various contributing factors including radiation. It was thought, however, that radiation, although a factor, was a minor one. The Laboratory did collect a little data on the approximate effect of radiation to walls which indicated that it was not of great importance.

The mere fact that this factor, which was considered small and relatively unimportant ten years ago, comes up now as a major factor, is, I believe, a measure of the progress we are making. As we eliminate the more important factors that affect us in our data on heating and ventilation, the factors which are of less importance to start out with become relatively of greater importance and we have to attack them. In other words, the measurements that could be made with a foot rule ten years ago have been made and the remaining measurements now have to be made with a micrometer. We are working to a higher degree of accuracy.

In Mr. Close's discussion of this paper he quoted quite largely from the work of Aldridge on heat dissipated from the body by radiation. I think that subject needs further investigation for in my opinion radiation is not quite as important a factor as brought out by Aldridge. This should be true for several reasons. Any considerable part of our body does not radiate from the nude surface at or near blood temperature, but rather from the temperature of the outside of our clothing which is much nearer the temperature of the air than that of the blood. For instance, with a body temperature of 98 F and an air temperature of 70 F, our outside clothing surface temperature is but a few degrees higher than the air. As a result small variations in the temperature of wall surfaces to which we radiate become relatively of less importance.

There were quoted, effects reported by earlier investigators showing that with a lowering of the wall surface temperatures to which we radiate by 20 deg, the rate of heat dissipated from the body surface by radiation was double. While the fact that these data are for a nude person is mentioned by Mr. Close, it should be emphasized more strongly for no such increase in heat loss would result for a normally clothed person.

Taking the Laboratory data on heat emission from the body for various atmospheric conditions, we find that if the temperature of surrounding wall surfaces and also the air temperature falls 10 deg, the total heat loss by both radiation and convection increases only about 5 per cent. However, this 10 deg fall in surrounding temperatures is also accompanied by a compensating decrease in heat loss by evaporation of 5 per cent. That compensation, however, only takes place for a drop of about 10 deg. After we get down to 60-deg air and wall temperatures, the heat loss by evaporation becomes a minimum and we have little control of heat loss and our heat production mechanism is called upon to supply any additional loss. While there is no question but what radiation to surrounding walls is theoretically a factor in comfort and should be investigated by the Laboratory, it is quite probable that the effect will not be very great.

I would predict that for the variation in temperature met with in wall surfaces for reasonable variations in wall construction the effect on the effective temperature will be no more than a couple of degrees. If this is found to be the case it can be taken care of in practice by saying that for walls having a

conductance above a certain value and in zero weather, the effective temperature maintained should not be that shown by the comfort line in our present chart but by a slightly higher temperature.

MR. CARRIER: When I said temperature, I meant temperature of a still thermometer both the wet-and the dry-bulb and if you sling the thermometers you will get a different result that will not be representative. In other words, to integrate that you have got to have a still thermometer similar to a still body and if you use a still thermometer instead of having a couple of degrees difference, I do not think you will have any.

THORNTON LEWIS: First I would like to congratulate the authors on the paper. It is very apparent to all of us from the discussion that they have started something and it is going to bring around some very interesting data for the use of all scientists and heating engineers. I have no intention of discussing this matter in such a profound way as the scientists who have preceded me, but there is one observation that perhaps might throw a little light on this question of trying to duplicate the effect on the human body and that might help in some further investigations that Professors Kratz and Willard may make.

Last summer it was my pleasure to be able to visit the Waterford Research Station in England where the British Government is conducting research relating to heat. You will all remember that panel heating is in vogue in England. In this station, and in the method of heating that most English engineers today are using, they are attempting to heat almost entirely by radiation and to minimize any heating effect from convection—to eliminate it, if possible. At the Waterford Station, that is just exactly what they are trying to do, but they recognize that in heating principally by radiation, a thermometer or a thermocouple would undoubtedly not register the true effect of the human body. So they built a copper cylinder, worked out mathematically, so that the surface of this copper cylinder approximated the surface of the human body that receives heat or heat effects by radiation, and then they used a thermocouple to measure the effect on this cylinder.

The report on that work is available and I would just like to suggest, in connection with the further work that Professor Kratz is doing, that it might be quite interesting. It would answer the point that Mr. Carrier has brought out of more correctly evaluating the effect on the human body of radiation as well as air temperature.

P. J. MARSHALL: It is apparent with present design of heating systems to maintain 70 F at the 5-ft breathing line temperature as indicated by the dry-bulb thermometer located on the wall at this distance above the floor is a very unreliable index of comfort conditions. You, of course, all know that the dry-bulb thermometer is not the true index of comfort conditions. We must take into consideration the wet-bulb, air motion and now this radiation effect. The information given in this paper accounts probably for the reason why many people contract colds when they stretch out on the davenport in the living room after an evening meal. As the living room in the average type of building construction has two and sometimes three exposed walls and a cold floor, they are subject to much colder conditions due to the low breathing line and radiation effect of cold walls and floors than at the breathing line. It would also be interesting to know what effect a slight pressure in the room might have towards maintaining a more comfortable condition.

Up to the time of the presentation of this paper I had a feeling that I was either above or below normal. Where conditions of temperature, humidity and air motion gave us an effective temperature of 65 deg, I very often felt decidedly uncomfortable. I refer particularly to the time when we held our convention in the Exposition Hall in Philadelphia. They had an effective temperature of 65 deg and I was decidedly uncomfortable. That perhaps can be now ascribed to the fact that my heat loss by radiation was excessive due to a cold concrete floor.

J. C. MILES: This subject of wall surface temperature is one that interests us most in the practical field. As the President said yesterday an engineer was just one jump ahead of a scientist and I, being in the practical field, might be one jump back of an engineer, but I do know this, that this question is highly important out in the field and I am glad they have done as much on it as they have.

Mr. Houghten spoke of the question being regarded as a minor factor for ten years and now being regarded as a major factor. In my estimation, I think the whole thing may be attributed to the women's wearing apparel. There has been quite a change in the past ten years. The women seem to have sense enough to put on the right kind of clothes so that they get the proper body respiration. It is very important to health and comfort; there is no question about that. A difference of 5 deg in surface wall temperature is going to make a lot of difference in the comfort on very thin, silk stockings, whereas it may not make any difference on the outside of this coat, as Director Houghten mentioned.

I was in St. Louis last winter and a woman complained about her room being cold and I did not think it was. She said, "I am not going to let you go away from this house thinking I am a crabby old lady." She took hold of my coat and said, "Take that off and sit down here for five minutes and find out what I have to experience." There was a cold wall about 2 ft to the right of this woman and the wall did cause a cool feeling.

There is a point that I think has not been taken up here and I hope Professors Kratz and Willard will do something about it. I notice that this paper is based on the condition of still air inside buildings, or inside the room. We may be able to raise the wall surface temperature by convection with air motion and in view of the fact that temperature, humidity and air motion are the essential requisites to comfort, I hope that Professors Kratz and Willard will find some means whereby they can find out the effect of inside air movement.

F. D. MENSING: I do not think we pay enough attention to the temperature of the floor, what goes underneath the floor, and what the temperature is underneath the floor. The same would also apply to the ceiling. We are having troubles with certain types of radiation which theoretically are doing the work but practically are not. The cause, I suspect, is due to a condition in the floor.

An example will show you how accurate the human being is. It was necessary to ventilate a room in an airplane factory where they applied a material with a spray. The air change in that took place every 2½ min, theoretically. Smoke bombs were turned loose at the point where the air entered, voided into the room and then the room cleared. From the time the bomb was lighted until you could see no evidences of smoke was 13 min. Remember this is an airplane factory, where we had to bring in the complete machine. This apparatus had to

be moved around on the floor. The men complained that they were uncomfortable, could not stand it. A line of thermocouples was hung and they all registered 70 F. That is the temperature we worked for, but it was not comfortable. We then hung a thermostat from the ceiling, as low down as we could. We got the owners to agree to not use that section of the floor; in other words, place it in the center of the work area; and all our troubles disappeared. These human beings were more accurate than our thermometers. Comfort does enter into industry. Take an operator working on a machine; it is the same story; the room may be 70 F but if the machine is 60 F, you are not going to get production.

PROFESSOR KRATZ: When we started out to write this paper we had measured just the surface temperature on the inside surface of the wall. We realized that this was a factor that might have some bearing on the human comfort but had no intention of going into an elaborate determination of all of the factors affecting human comfort nor of repeating any of the work that had been done to establish standards for measuring and defining human comfort. All that we were concentrating our attention on was just this one factor, the temperature at the inside surface of the wall, and by calling attention to the fact that this was one factor in a very complex problem we thought we might be able to promote some discussion. Apparently we have at least accomplished that result, but I cannot hope to answer all the discussion that has been given here. The problem is very complex. It may be a simple practical solution or it may be taken care of mathematically, but the ultimate solution will not be obtained by neglecting or overlooking any of the factors, and that has been the main purpose of this paper: to call attention to this one factor and to promote discussion on that one phase.

L. A. HARDING: The paper deserves very careful consideration and should evoke wide discussion among the members as the point raised has a bearing on future research dealing with the problem of comfort. This paper forcibly calls attention to a fact that is very frequently observed in home heating, namely that the occupants do not always feel comfortable when a 70-F room dry-bulb temperature is being maintained in cold weather. The effect is more often attributed, perhaps erroneously, to carrying a very low relative humidity necessitated to prevent moisture precipitation on single glass.

The statement was made by the author of a recent paper, describing panel heating, that a person normally clothed will feel perfectly comfortable in a room air temperature of 60 F, when the inside wall surface temperature is maintained at 75 F.

The layman naturally interprets the condition of the air as to temperature and relative humidity within the comfort zone of the A.S.H.V.E. comfort chart to mean that the conditions shown will produce a state of well-being or quiet enjoyment called comfort.

The engineer interprets the chart values to mean equal degree of comfort on any effective temperature line. It would appear however that either interpretation may be upset by changes in the wall surface temperature to values quite different from the surface temperatures maintained during the tests when the comfort chart values were determined.

It may be assumed that, with the first condition mentioned, in the second paragraph, a person is supposed to be comfortable, as the chart indicates, but

apparently this is not always the case. With the second condition, as mentioned in the fourth paragraph, he is not supposed to be comfortable, according to the chart, but to his surprise he finds that he is comfortable.

Comfort, as applied to a normally healthy person at rest, requires that the body temperature remain practically constant. This condition, as is well-known, can only exist when a fairly exact heat balance is maintained, between the heat liberated within the body due to various chemical reactions and the heat given off by the body. It is assumed that the air supplied is from an uncontaminated source.

The function of clothing and a heating system are identical, that of so regulating the heat dissipated to the surroundings that the heat balance is maintained. A combination of these two items is evidently essential for convenience and modesty.

It is generally assumed that the heat dissipated by a normally clothed person at rest in still air is divided into the following approximate percentages: 30 per cent by convection, 43 per cent by radiation to colder surfaces, 24 per cent by evaporation from the lungs and the remainder, or 3 per cent by other causes when the air temperature is maintained at 70 F with 25 per cent relative humidity. It would seem necessary to further assume that the temperature of the wall surfaces are somewhat less than the air temperature. It may be rightfully assumed that the percentages stated, if they are correct, also represent, for each item mentioned, the percentage that the item reflects in contributing to our comfort.

The convection loss in still air depends upon the surface coefficient of the clothing and the temperature difference between this surface and the room air. The radiation loss depends upon the temperature difference between the clothing and the surrounding wall surface temperature and its coefficient of absorption.

The A.S.H.V.E. Comfort Chart does not take into account the effect of body radiation loss as affected by variations in wall surface temperature. As this loss ordinarily constitutes the greatest percentage of any of the items mentioned, it would appear necessary to accompany this chart with a statement to the effect that the wall surface temperature should not be less than a certain specified amount when this chart is to be applied, as a guide to the engineer for residence heating.

It may be found that the required wall surface temperature is higher than exists with the usual wall constructions employed in cold climates. Under this condition the only alternatives for the occupant would be to increase the body insulation in order to decrease the body radiation loss, or to increase the room air temperature above the chart values in order to decrease the convection loss.

It is, I believe, essential to run a series of comfort tests, employing both male and female subjects clothed with winter garments, maintaining the inside wall surface temperatures within the temperature range as found in practice.

The present comfort chart is no doubt correct for use with assembly halls and theatres, as in this case only those occupying seats adjacent to outside walls would be affected by radiation to colder wall surfaces, but some doubt may be expressed as to its value or correctness for heating and air conditioning of a residence.

No. 872

CAPACITY OF DRY RETURN MAINS FOR STEAM AND VAPOR HEATING SYSTEMS

By F. C. Houghten[1] (*MEMBER*), and Carl Gutberlet[2] (*NON-MEMBER*),
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THE capacity of pipe for carrying steam, condensate and air in various parts of steam heating systems has been under investigation at the A.S.H.V.E. Research Laboratory, and at Carnegie Institute of Technology, in cooperation with the Research Laboratory since 1922. A number of publications[3] have resulted from this study, giving the capacity of pipe for various parts of the steam supply side of heating systems. The practical application of these results has been made use of in the pipe sizes given in Chapter 20 of The A.S.H.V.E. Guide, 1930.

Upon completion of the study of the supply side of steam heating systems a little over a year ago, the Society was asked to supply data on capacity of returns. The work on this study has since been carried on by the Laboratory staff in the heating and ventilating laboratory, Carnegie Institute of Technology, where suitable space for a test set-up, including a 22-ft return riser and a 540-ft dry return main, was available. This paper gives the results of tests on a 1-in. dry return main. Data were collected simultaneously on the capacity of return risers, but this phase of the study is not yet completed.

Test Set-up and Operation

The test set-up used for the investigation is shown in Fig. 1. The pipe and other equipment, however, were actually arranged other than shown.

Steam was supplied from a high pressure main *A*, through a suitable arrangement of throttling, by-pass, and pressure reducing valves to a 7-ft section of 8-in. pipe *B*, so as to maintain the uniform pressure desired. From *B*, steam was taken through the 4-in. riser and distributing main *C, D, E,* to a battery of six unit heaters, *F* to *G*. The condensate from the unit heaters was collected in the 1-in. horizontal return *HI*, and was returned through the riser and 1-in. return main, *IJK* to the gravity receiving bucket *L*, or to the vacuum pump,

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[3] A. S. H. V. E. Transactions and Journal, 1922 to date.

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where the non-condensable gases were liberated to the atmosphere and the water delivered to either of the two receiving tanks M or N.

The unit heaters were of several makes ranging in size from 2,200 to 150 equivalent square feet of direct radiation. The four larger units were drained through float-type blast traps with thermostatic by-passes for air. These connections are shown in Fig. 2. The two smaller units were drained through

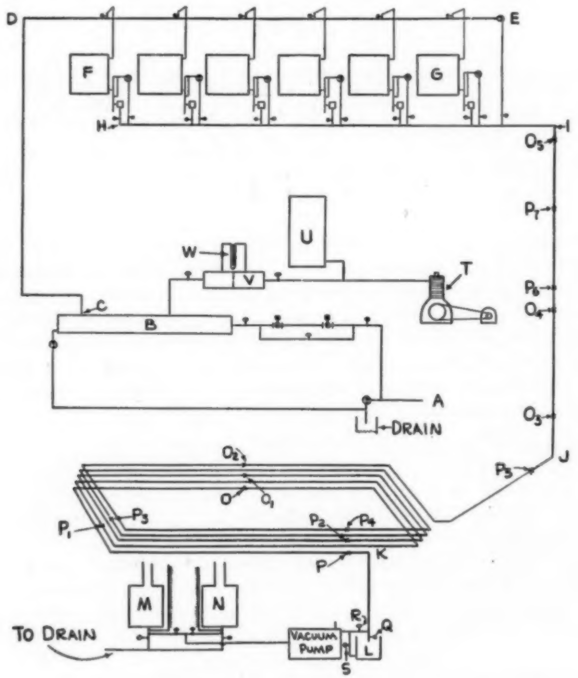

FIG. 1. LABORATORY SET-UP FOR STUDYING CAPACITIES OF DRY RETURN MAINS

thermostatic traps as shown in Fig. 3. A water column gage glass was connected to indicate any water accumulation in the connection between the unit and the blast trap, Fig. 2, or the thermostatic trap, Fig. 3. Any of the units could be turned on at will and the fan speed of one of the larger units could be varied. With this arrangement any rate of steam condensation desired could be obtained and controlled.

Radiators of 28 equivalent square feet capacity were supplied with steam and drained into the return at points O, O_1, O_2, etc. The connections of these radiators are shown in Fig. 4. At points P, P_1, P_2, etc., along the return, additional

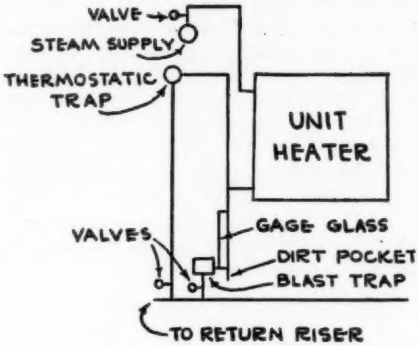

Fig. 2. Detail Connection of Large Unit Heaters

provision was made for measuring the pressure in the return main besides at those points at which radiators were located. The arrangement for measuring the pressure in the riser is shown in Fig. 5. The arrangements for measuring the pressure in the return main are shown in Fig. 6.

Pressures were measured at points *P* and *O* along the return in order to indicate the pressure in the return at these points and by difference the pressure drop along the line for any condition of flow. The radiators were used in order to determine whether or not the return was overloaded at these points. If upon turning steam into the radiators the air was eliminated and the radiator heated up quickly, the return was considered not to be overloaded. On the other hand, if the radiator did not heat satisfactorily, the return was overloaded for the particular vacuum applied at the pump.

In the radiator connection, Fig. 4, a valve was placed between the thermostatic trap and the return so as to insure against water backing up into the radiator should the return be overloaded. In order to make a determination of the pres-

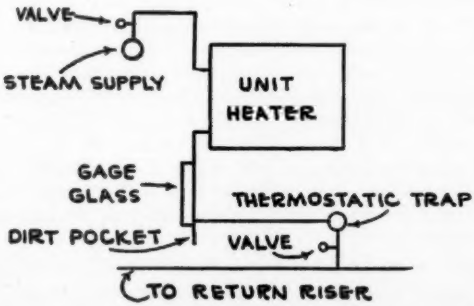

Fig. 3. Detail Connection of Small Unit Heaters

sure in the return, at this point, the valve between the thermostatic trap and the return was opened and when the pressure in the radiator had adjusted itself to that in the return it was read on the manometer. Observation was also made of any water accumulation in the radiator as indicated in the gage glass. The steam supply valve of the radiator was then turned on to give a pressure of steam entering the radiator of 1 oz above the pressure in the return as previously read in the radiator and the rate of heating up was observed. Observation was also made during the heating up period of any water accumulation in the radiator as indicated by the gage glass or any other peculiarity of operation.

The arrangement in Fig. 5 for measuring pressure in the riser was used in order to guard against the water backing up into the manometer. The pipe connection between the 2-in. nipple and the return was made large enough that

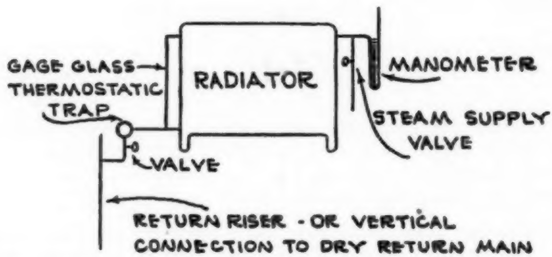

FIG. 4. DETAIL OF RADIATOR CONNECTIONS USED AS INDICATOR OF OVERLOADED RETURNS

water would not seal across it, but would drain out freely. The arrangement shown in Fig. 6 for measuring the pressure in the return main also insured against water backing up in the manometer or sealing across the pipe connection.

The system could be operated either by gravity return or by vacuum pump return. By opening valve *Q*, and closing valve *R*, the condensate returned by gravity, draining into the receiver *L* and by opening valve *S* the water contained in the receiver could be transferred as collected into the measuring tanks *M* and *N*. By closing valves *Q* and *S* and opening valve *R*, the system operated by vacuum pump return, the pump delivering the water directly to the measuring tanks. The measuring tanks were barrels of approximately 50-gal capacity. Each tank was supplied with an upright extension above the barrel made by flanging on a two-foot section of 4-in. pipe. Each barrel was fitted with a gage glass so that the water level in the barrel or smaller chamber at the top could be determined easily. For accurate determinations of return condensate the measurements were always made with the water in the smaller constriction above the barrel so that a small error in the water level had little effect on the accuracy of results. Either measuring tank could be filled or drained at any time by operating the valves.

Soon after starting the study it was realized that the set-up described would give satisfactory data on the capacity pressure drop relationship for the return when only the non-condensible gases contained in the steam were being handled,

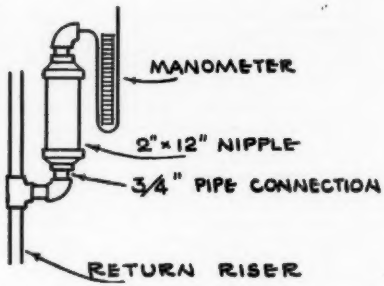

FIG. 5. CONNECTION OF MANOMETER FOR MEASURING PRESSURE IN RETURN RISER

excepting any small amount of air eliminated from the units or leaking into the system due to the vacuum. Since the amount of air handled by the system under these conditions was extremely small, the results obtained gave practically the ultimate capacity of the returns for handling condensate with a very minimum amount of air and the extremely high capacities found for these conditions demonstrated clearly that return pipe sizes for practical use must not be based upon these high values.

This naturally brings up the question of how much air the returns of a steam heating system should handle. Widely different estimates of this value may be made based upon various assumptions of the relative rate of air expulsion from cold radiators and of steam condensation. In a vacuum pump system, the problem is further complicated by the probability of air leaking into the system through loose joints and fittings in poor installations.

Soon after starting the investigation it was realized that in order to base satisfactory tables for determining return pipe sizes on the results of the investigation, data must be established giving the capacity of the return for various rates of handling condensate and air. In order to make this possible, means were provided for supplying air into the system as previously described.

This was done by supplying air to the main *B*, Fig. 1, through a calibrated

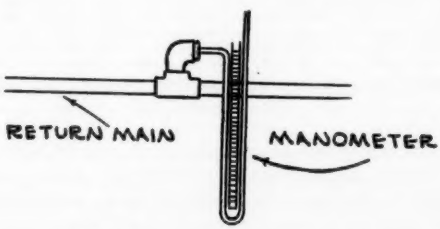

FIG. 6. CONNECTION OF MANOMETER FOR MEASURING PRESSURE IN RETURN MAIN

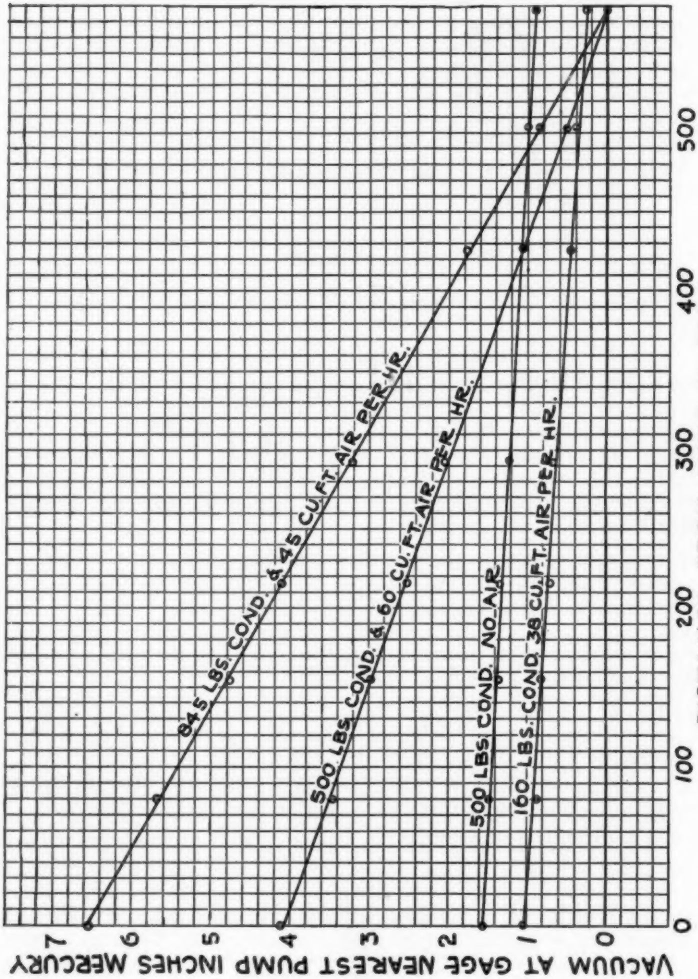

FIG. 7. PRESSURE DROP THROUGH THE DRY RETURN MAIN FOR VARIOUS AIR AND CONDENSATION LOADS

orifice *V*, from the motor driven air compressor *T*, and auxiliary storage tank *U*. With this arrangement the observer controlling the steam pressure in *B* could also control the rate of air supplied as indicated by the orifice meter *W*. The air admitted into the main was heated to steam temperature by the steam before passing through the units to the return side of the system.

In making a test under anv specified condition of operation the steam pressure

in the main *B* and the rate of air supply were regulated and held constant. The right number of unit heaters were turned on and the speed of the fan on the unit subject to regulation was adjusted to give the condensation rate desired. These conditions were then maintained until equilibrium of flow was established, after which test data were collected. During the test period the rate of condensation return as measured in the tanks was recorded and at the same time observations were made of the pressures at the various locations *O* and *P* along the return riser and main. The small radiators O, O_5 located along the return line were then turned on with a pressure on the steam supply end of the radiator one ounce above the pressure recorded in the return at this point. The time required for eliminating air from the radiator through the return and entirely heating the radiator was observed. Any test in which a period of more than 20 min

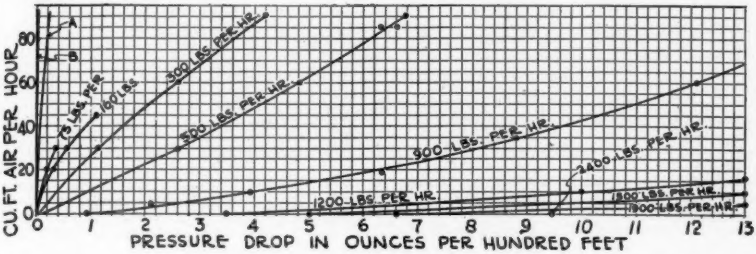

FIG. 8. RELATION BETWEEN PRESSURE DROP IN THE DRY RETURN MAIN AND AIR CARRIED FOR VARIOUS CONDENSATION RATES

was required for eliminating air from and heating up the 28-sq ft radiators was not used.

EXPERIMENTAL RESULTS

The pressure drops observed throughout the length of the 1-in. dry return main for several representative rates of condensation, air flow, and required vacuum at the pump are given in Fig. 7. From these curves the pressure drop in ounces per 100 ft length of run were calculated for use in determining the pressure drop, condensation and air flow relationships desired.

In all of the tests for any given condensation and air flow rates, the vacuum at the pump was adjusted approximately to allow the farthest radiator from the pump to heat up in about 15 min. It was soon found that the time required for the radiator to heat up bore a relationship to the pressure in the return at the point where the radiator drained into it. Whenever the pressure in the return reached 2 oz above the atmosphere, the radiator would require an excessive time for heating regardless of the fact that the steam pressure at the entrance to the radiator was adjusted to one ounce above the pressure in the return.

The relation between pressure drop in the main in ounces per 100 ft length of run and the amount of air handled through the main in cubic feet per hour is given in Fig. 8 for several rates of condensation flow in a 1-in. dry return main.

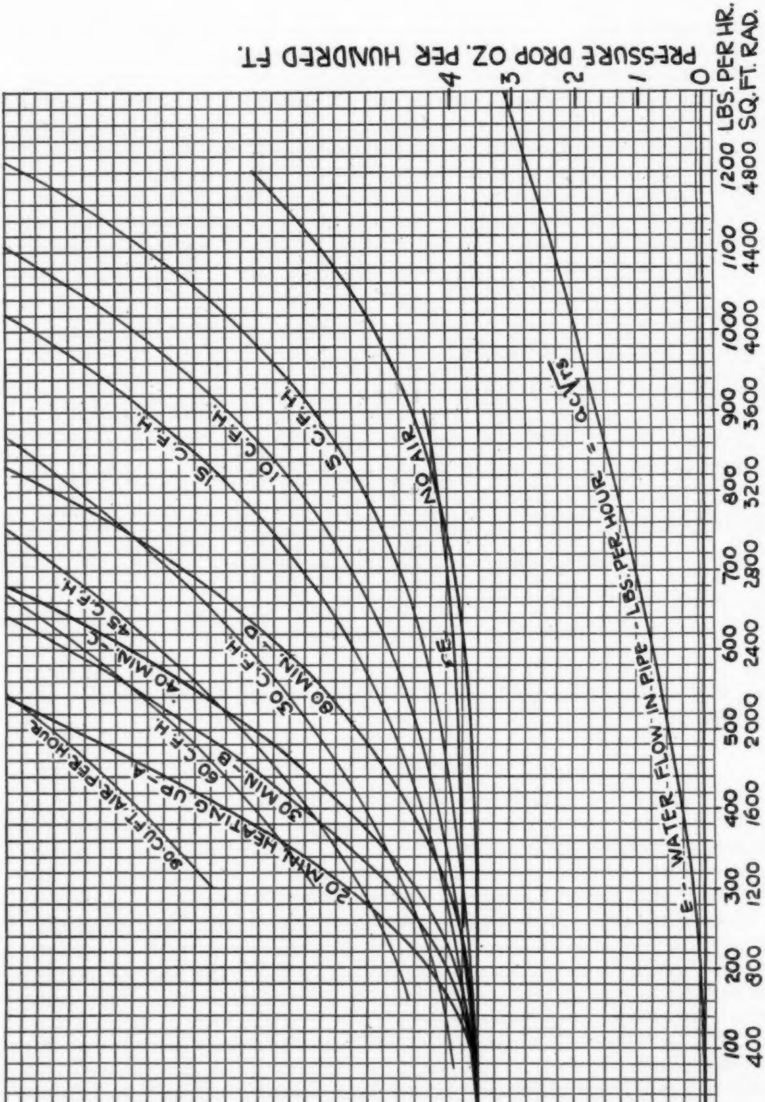

FIG. 9. RELATION BETWEEN PRESSURE DROP AND CONDENSATION LOAD FOR VARIOUS RATES OF AIR ELIMINATION THROUGH THE DRY RETURN MAIN

The points on the various curves were each obtained by averaging the pressure drops obtained from curves of individual tests as plotted in Fig. 7. The curves for loads ranging from 75 to 900 lb of condensation per hour were each determined by a large number of tests made under most careful conditions. The curves for condensation rates from 1,200 to 1,900 lb per hour are of relatively less practical value and these curves were determined by comparatively few tests under less favorable conditions. The limit in the quantity of air handled in the tests with loads of 75 and 160 lb condensation per hour was fixed by the satisfactory operation of the system. Curve *A* gives the capacity of a a dry 1-in. pipe for carrying air only as reported by O'Bannon.[4] Curve *B* gives similar data calculated from Unwin's formula. These two curves are of value in showing the degree to which the condensation carried by the main reduces the capacity of the dry pipe for carrying air only.

The relation between rate of condensation in pounds per hour or equivalent square feet of radiation based upon ¼ lb per square foot, and pressure drop in the 1-in. dry return main in ounces per 100 ft length of run, is given in Fig. 9 for various rates of handling air ranging from *no air added* to *90 cu ft air per hour.* These curves were developed from the curves for different loads, Fig. 8. From this chart may be derived the capacity of a return main for any desired rate of handling air or allowable pressure drop within reasonable limits.

The curves *A, B, C,* and *D* give the relationship between condensation load handled and pressure drop, based upon certain definite relationships between rate of air and condensation handled. These relationships were calculated to apply to various rates of heating up a cold system in which all condensation takes place in cast iron tubular radiation. As a result of measurement of the volumetric capacity of 15 tubular radiators of various sizes as sold by four of the larger manufacturers it was found that this type of radiation had an average capacity of 0.0142 cu ft per rated square foot of radiation. Assuming that air must be eliminated in a period of 20 min from an entire system containing all tubular cast iron radiation, air must be eliminated from the system at the rate of 3×0.0142 or 0.0426 cu ft of air per hour per square foot of radiation or 0.170 cu ft of air per pound of condensation handled. Curve *A* is based upon the limitation of air at this rate.

There is doubt regarding the justification of the assumption of this rate of air elimination. It has been suggested that to the air elimination from the radiators should be added the air elimination from the supply and return pipe. It does not seem logical, however, to add the air eliminated from the supply piping for the reason that an equivalent volume of air should be driven through the radiator and out through the return before the return begins to handle condensation, and as shown by curves *A* and *B,* Fig. 8, the capacities of the returns for handling air alone are extremely large compared with their capacity for handling air and condensation. Hence the elimination of air from the supply piping should be of little consequence, and since the returns are comparatively small it may likewise be assumed that the elimination of air from the return side of the system is no great factor.

It is probable that the tax on any system in eliminating air during the heating

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[4] Simultaneous Flow of Water and Air in Pipes, by L. S. O'Bannon, A. S. H. V. E. TRANSACTIONS, Vol. 30, 1924, p. 157.

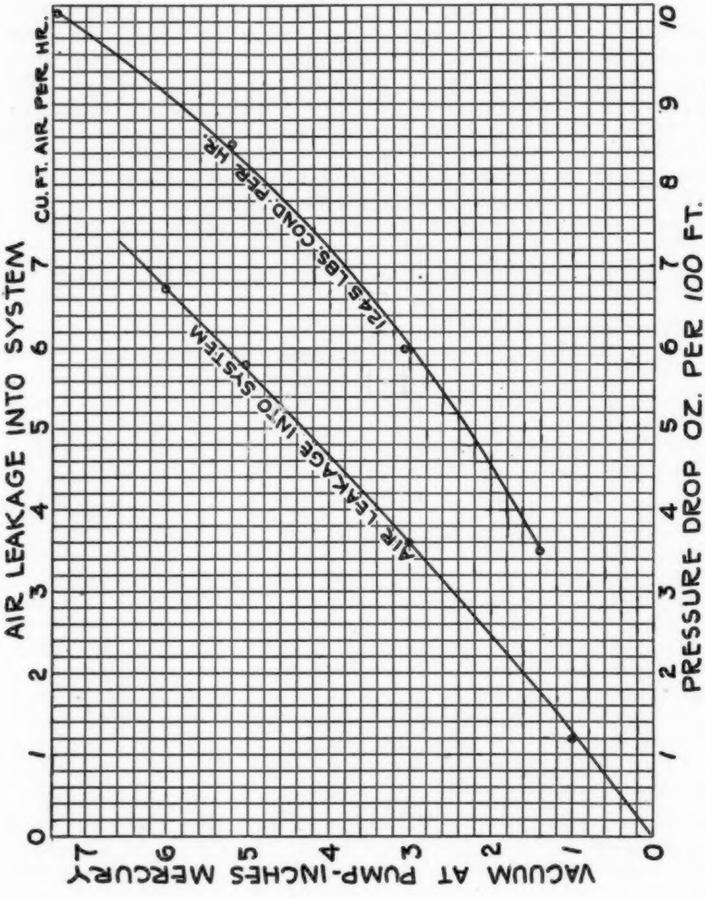

FIG. 10. RELATION BETWEEN VACUUM AT THE PUMP AND PRESSURE DROP ALONG AND LEAKAGE INTO THE RETURN SIDE OF THE SYSTEM

up period, even though the entire system is heated up at the same time, is much less than that indicated by curve *A*, for the reason that while the first portion of the radiators is being heated up, air from these portions is being eliminated through returns carrying much less than the indicated weight of condensate; hence the greater ease with which the system will be freed from air.

In many cases, it is not necessary to heat a system up in as short a period as 20 min and in comparatively few cases is it required that the entire system be heated up during the same time at this rate. Hence it would appear that curve *A* indicates relationships for air elimination at rates much higher than need usually be considered.

Curves *B, C,* and *D* indicate rates of air elimination of 66, 50 and 25 per cent of that indicated in curve *A,* or curves *B, C,* and *D* give the condensation pressure drop relationships for heating up periods of 30, 40, and 80 min, respectively.

The curve designated as *no air* gives the relationships for operating the experimental system when no air was being added to the system. This curve gives the relationship found when air was being eliminated from one at a time of the small radiators located along the return and draining into it as a measure of whether or not the return was overloaded. It also carried the small amount of air leaking into the system due to the reduced pressure maintained by the vacuum pump and any non-condensable gases in the steam.

The non-condensable gas in the steam supplied to the test system was very small and found by measurement to be 0.00018 cu ft of non-condensable vapor per pound of condensation. The rate of air leakage into the system due to the vacuum maintained is given in Fig. 10. This leakage was found when the supply valves to the unit heaters were closed off and the indicated vacuums were applied to the return and the unit heaters. Under operating test conditions the actual in-leakage would be considerably less than these values for the reason that under these conditions much lower pressures than maintained by the pump existed in other parts of the system. While these facts indicate that a very small quantity of air was actually handled in obtaining the data shown in the curve *no air,* Fig. 9, this curve represents conditions where the system was actually handling some air and non-condensable vapors and does not represent a true condition of water flow in a filled pipe.

In order to indicate the relation between the maximum condensation carried for any pressure drop as indicated by curve *no air* and the capacity of a similar pipe flowing full of water, and the curve *E* was plotted based upon the following formula by Darcy:

$$Q = AC \sqrt{rs}$$

where $Q =$ pounds water per hour flowing through the pipe.

$A =$ area of the pipe

$r =$ the mean hydraulic radius.

$s =$ the pressure drop

$C =$ a constant.

The zero of this curve is dropped down to 3.61 oz per 100 ft length of run below the zero of the test curves; the value of 3.61 being the head produced in a 100 ft length of pipe filled with water and having a pitch of 1 in. in 16 ft. The pressure drops indicated for the various curves in Figs. 7, 8, and 9 are based upon the difference in manometer readings giving the pressure differences between the interior of the main and the atmosphere at the various points of measurement along the line and these pressure drops do not include any head which might be created by the pitch of the pipe. If the pressure drops indicated in the test curves (Fig. 9) are the result solely of air flow in a space in the upper portion of the pipe above the water level of the returning condensate then the relative location of the zero pressure ordinates of the curve *E* and curve *no air* is correct. Otherwise it is not.

Curve *F* gives the relationship between the rate of water and air handled and

pressure drop in a 1-in. test pipe as reported by O'Bannon. This curve is for a rate of handling air of 60 cu ft per hour. It shows a much lower pressure drop for the same rate of handling air than indicated by the similar curve established in this investigation. The reason for this becomes apparent when the different conditions under which the data were collected in the two investigations are compared. Curve *F* is for quiet, non-turbulent flow of water in the lower portion of a pipe with air flowing above. The test curves resulting from this investigation are for a condition of more or less turbulence as found in the actual test system.

In order to observe the degree of turbulence found in the flow, a 3-ft section of glass with 1.04 in. internal diameter was inserted in the dry return main immediately below the return riser. Under no condition of flow as regards rate of condensation or air handled was the flow of condensation found to be steady. On the contrary, it was found to flow in waves, filling a considerable portion of the pipe and occasionally all of it.

For condensation rates of 300 lb per hour and higher the pipe was filled by waves as often as once per 2 min. When no air was being added the flow was less turbulent and when the pipe was filled it was by a slug of water rather than a short wave as was the case when large volumes of air were being handled. For loads of 150 lb per hour and less the waves seldom reached the top of the tube. For no air added with these loads the water surface was little disturbed while with greater rates of air flow the surface was thrown into waves.

The turbulence in the pipe was probably caused by the disturbance incident upon the entrance of the condensate and air from the riser and also to intermittent passage of water and air through the traps.

Early in the investigation it was found that the pressure drop for a given rate of condensation flow varied over rather wide limits depending upon the vacuum applied to the return side of the system. This was particularly true for large condensation loads when small amounts of air were being handled. Fig. 10 shows the relation between vacuum applied and the pressure drop through the length of the return main.

It was first thought that the variation in pressure was due to air leaking into the system and as a result measurement of this leakage was made as previously stated and plotted in this figure. By adding additional air and measuring the increase in pressure drop it was shown that the air leakage could not account for the increase in pressure drop with increased vacuum at the pump shown in the curve. It was found, however, by computation that re-evaporation of condensation upon passing from the higher pressure on the unit heater side of the trap to the lower pressure maintained by the vacuum on the return side of the trap might account for the variation in pressure drop with vacuum.

A reduction in pressure of 1 in. in mercury results in the liberation of 3.5 Btu per pound of condensate at a steam temperature of 215 F. This is sufficient to re-evaporate $\frac{3.5}{970 \times 0.03802}$ cu ft of steam per pound of condensation or 42 cu ft of steam per hour for a condensation rate of 900 lb per hour. Assuming that steam in the return had the same effect as air on pressure drop for a given rate of water flow, 42 cu ft of steam per hour with a 900-lb load would be found

from Fig. 9 to increase the pressure drop from one to approximately 9 oz or 8 oz per 100 ft length of run.

It is of course true that all of this liberated heat will not be available for re-evaporation because of heat dissipation from the piping between the unit heaters and the traps and, further, the amount of steam formed will rapidly decrease by condensation due to heat dissipation along the return . However, it is probable that re-evaporation accounts in a large measure for the variation in pressure drop with vacuum maintained for a given condensation load as shown in Fig. 10.

The limitation of condensation and air loads, for gravity return is given in Fig. 11. Curves *A, B, C, D* give rates of air elimination corresponding to 20 30, 40 and 80-min heating-up periods respectively.

Practical Application of the Data

The relationships given in the curves Fig. 9 to 11 may be used for developing tables giving capacities of 1-in. dry return mains. The acceptance of such tabular values must, however, be based upon certain assumptions, including the allowable pressure drop, the rate of heating up or air elimination and any factor of safety desired for covering other contingent factors. These assumptions can not be determined by laboratory test but must be based upon experience gained through practice and judgment.

Table 1 gives the capacity of a 1-in. dry return main for various allowable pressure drops in ounces per 100 ft length of run and various rates of air elim-

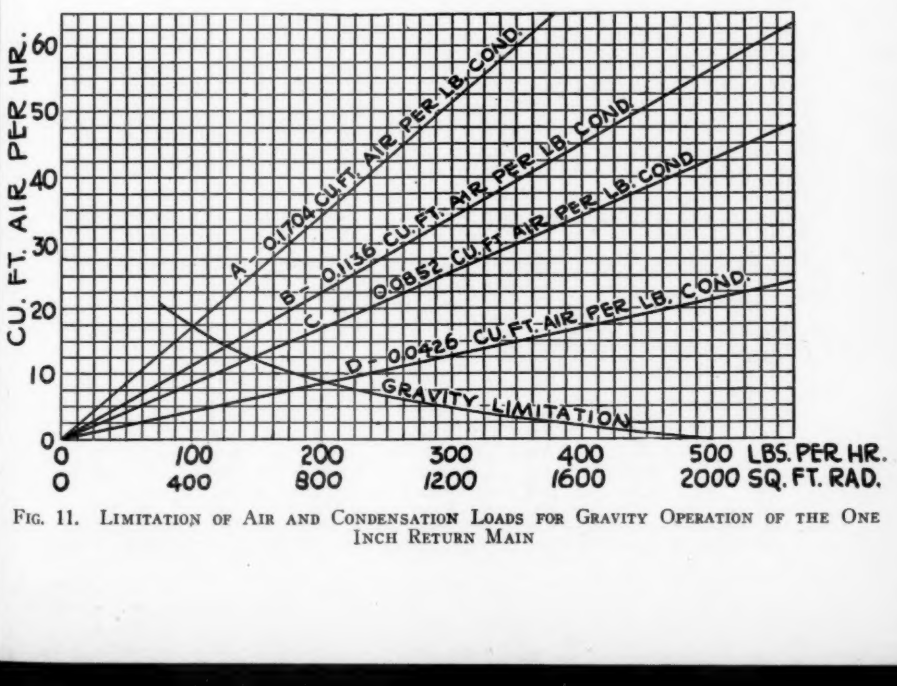

Fig. 11. Limitation of Air and Condensation Loads for Gravity Operation of the One Inch Return Main

TABLE 1. CAPACITY OF 1-IN. DRY RETURN MAINS IN SQ FT OF EQUIVALENT RADIATION FOR VARIOUS RATES OF AIR ELIMINATION. (EXPERIMENTAL RESULTS)

Pressure Drop—Oz Per 100 Ft	Period of Air Elimination				
	20 min	30 min	40 min	80 min	No Air Elimination
¼	512	586	684	940	2480
½	672	800	930	1160	2964
1	888	1060	1208	1560	3560
2	1180	1396	1568	2034	4200
3	1410	1648	1856	2368	4608
4	1600	1892	2096	2650	4936

ination based upon the elimination of 0.0142 cu ft of air per square foot of radiation during the heating-up periods.

In the application of the laboratory test data for sizing pipe for the supply side of steam heating system, practical application has dictated the use of a 20 per cent factor of safety to cover variations in pipe capacity due to variations in pipe size, smoothness of pipe, constrictions due to improper cutting and other contingent factors.

Table 2 was developed from Table 1 by the application of a 20 per cent factor of safety. This is offered for comparison and is not necessarily recommended for adoption at this time.

Table 3 gives the capacity under which the system was found to operate satisfactorily with gravity together with the maximum rate at which air could be eliminated from a system of all tubular cast iron radiators. Table 4 is developed from Table 3 by the application of a 20 per cent factor of safety.

It is of interest to compare some of the values given in Tables 1-4 with capacities of dry return mains as found in the pipe size tables in THE GUIDE 1930. THE GUIDE gives capacities of dry return mains in vacuum return jobs as given in the last column of Table 2. These are seen to compare very closely to the results of this investigation for a 20-min rate of air elimination, after the application of a 20 per cent factor of safety as given in the second column of the table.

TABLE 2. CAPACITY OF 1-IN. DRY RETURN MAINS IN SQ FT OF EQUIVALENT RADIATION FOR VARIOUS RATES OF AIR ELIMINATION. (EXPERIMENTAL RESULTS LESS 20 PER CENT FACTOR OF SAFETY.)

Pressure Oz per 100 ft	Heating Up Period of					From GUIDE 1930[a]
	20 min	30 min	40 min	60 min	No Air Elimination	
¼	409	468	547	752	1984	
½	537	640	744	928	2371	
1	710	848	966	1248	2848	700
2	944	1116	1254	1627	3360	994
3	1128	1318	1484	1894	3686	
4	1280	1513	1676	2120	3948	1400

a Capacities of dry return mains for vacuum installations. No rate of air elimination given.

TABLE 3. CAPACITY OF GRAVITY DRY RETURN MAINS IN SQUARE FEET OF RADIATION WITH CORRESPONDING MAXIMUM RATES OF AIR ELIMINATION. (EXPERIMENTAL RESULTS.)

Capacity sq ft Equivalent Radiation	Maximum Rate of Air Elimination	
	Time[a]	Cu ft per hour
200	7 min	25
400	20 min	17.5
600	45 min	12
1200	240 min	4.5
2000		0

[a] Time necessary for eliminating air from a system in which all steam is condensed in tubular cast-iron radiation having a volumetric capacity of 0.0142 cu ft per equivalent square foot of radiation.

THE GUIDE gives capacities of dry return mains in gravity return systems ranging from 284 to 460 sq ft equivalent radiation for pressure drops ranging from ½ to 4 oz per 100 ft length of run respectively. No rate of air elimination is given, however. Table 4 shows capacities, after allowing a factor of safety of 20 per cent, ranging up to 1,600 sq ft depending upon the rate of air elimination. The higher GUIDE value of 460 sq ft corresponds to a minimum time for complete air elimination of 25 min.

SUMMARY AND CONCLUSIONS

1. As a result of the investigation, curves in Fig. 9 give the relation between condensation carried by dry return mains pressure drop through the mains, and air carried.

2. The volumetric capacity of tubular cast iron radiation is given for the average tubular cast iron radiator and based upon these values the relation between capacity and pressure drops is given for various rates of elimination of air from the system.

3. It is shown in Fig. 8 that the capacity of dry return mains handling condensation even at comparatively low rates have very much smaller capacities for handling air with a given pressure drop than the same pipe handling air only. It is also shown that the capacity of a dry return main for carrying water and air is considerably lower than the capacity of the same sized pipe for handling water and air with quiet flow. It is shown that this is probably due to turbulent

TABLE 4. CAPACITY OF GRAVITY DRY RETURN MAINS IN SQ FT OF RADIATION WITH CORRESPONDING MAXIMUM RATES OF AIR ELIMINATION. (EXPERIMENTAL RESULTS LESS 20 PER CENT FACTOR OF SAFETY)

Capacity sq ft Equivalent Radiation	Maximum Rate of Air Elimination	
	Time[a]	Cu ft per hour
160	7 min	25
320	20 min	17.5
480	45 min	12
900	240 min	4.5
1600		0

[a] Time necessary for eliminating air from a system in which all steam is condensed in tubular cast-iron radiation having a volumetric capacity of 0.0142 cu ft per equivalent square foot of radiation.

flow in a dry return main caused by disturbances set up by entrance of condensate from risers and intermittent operation of traps.

4. The pressure drop in dry return mains increases with high vacuums maintained on the return side of the system for the same rate of handling condensation and air, probably caused by re-evaporation of condensation after passing through the traps.

5. Capacities of dry return mains are given in tabular form based upon laboratory results found and also by applying a factor of safety of 20 per cent to these results.

No. 873

LOSS OF HEAD IN SUBMERGED ORIFICES

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MEMBER

INTRODUCTION

IN designing pipe lines for water heating systems it is frequently necessary to increase the friction in a portion of the system in order to secure a balanced flow of water to the several radiators in the system. The additional friction head may be secured by inserting valves constructed so that the opening through the valve is adjustable and can be reduced sufficiently to secure the desired loss of head. The additional friction head may also be secured by inserting sections of pipes of sufficiently small size.

When neither of these two methods is to be used the additional friction head may be produced by contracting the cross section of the pipe in one or more places by inserting a union with a metal plate having an opening sufficiently small to secure the required loss of head.

Fig. 1 shows such a union with a plate having a central circular orifice. The stream of water flowing through the orifice will have a contracted section, as indicated in the figure. The head necessary to produce the increased velocity at the smallest section of the contracted stream will be (approximately) the loss of head caused by the orifice, because the energy stored in this increased velocity will be very largely transformed into heat.

If the degree of contraction of the stream after it has passed the orifice were known, it would be possible to calculate (with a fair degree of accuracy) the loss of head caused by the orifice. For example, if for the case shown in Fig. 1, the diameter of the pipe is 1.042 in., the diameter of the orifice 0.5 in., and the diameter of the contracted stream at the smallest cross section 0.4 in., as seems to be the case when the stream of water is discharging into air, then, for a velocity of 2 ips in the pipe, the velocity at the smallest cross section of the stream would be 13.6 ips, since the velocities are inversely proportional to the squares of the diameters. The head necessary to produce the increased energy stored in this increased velocity is

$$\frac{1.13^2 - 0.167^2}{64.4} \text{ or } 0.019 \text{ ft or } 228 \text{ milinches.}$$

It is not known, however, that the smallest cross section of the stream will be

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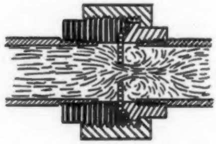

FIG. 1. UNION IN 1-IN. PIPE WITH DIAPHRAGM HAVING CENTRAL CIRCULAR ORIFICE AND SHOWING PROBABLE FLOW OF WATER THROUGH ORIFICE

0.4 in., nor is it known that the entire energy stored in the increase in velocity will be transformed into heat; a portion may be consumed in producing an increased pressure head in the section of the pipe beyond the orifice. In addition, a portion of the loss of head may be caused by friction along the perimeter of the orifice.

On account of these uncertainties it is necessary to make experimental determinations of the losses of head caused by orifices in order to have reliable data for use in the design of pipe systems. For example, in the case cited, experimental determinations (see Fig. 6) show that the loss of head is 150 milinches instead of 228, as indicated by the calculation.

DESCRIPTION OF APPARATUS AND TESTS

In order to make such a determination for diaphragm orifices in a 1-in. pipe, the apparatus shown in Figs. 2 and 3 was constructed at the Texas Engineering Experiment Station, in cooperation with the A. S. H. V. E. The pipe used was a good section of a new standard black pipe calibrated and found to be 1.042 in. in diameter. The two unions were standard black unions; the three piezometer rings, one of which is shown in Fig. 4, were of brass soldered to the pipe which was perforated with eight holes 0.067 in. in diameter and spaced 45 deg apart; the middle ring was used to remove air that might be trapped in the pipe between the two unions; the two outer rings were connected to the manometer tube and served to measure the loss of head in the pipe between the two outer rings. The connection of the piezometer rings to the manometer was made as shown in order to trap any air that might escape through the piezometer

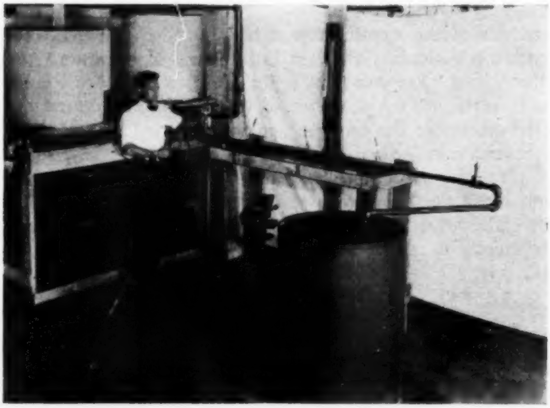

FIG. 2. APPARATUS FOR DETERMINING LOSS OF HEAD DIAPHRAGM ORIFICES

rings and prevent its entering the manometer tubing where it would affect the manometer reading.

The test was conducted in the usual manner. That is, water was maintained at a constant head in the supply tank and allowed to flow through the pipe system into the weighing tank. When the flow had attained a steady state, readings were taken from which the velocity of the water in the 1 in. pipe could be calculated and the friction head between the two outer piezometer rings recorded.

Results of Tests

The first two series of tests determined the friction head in two standard black unions and in 4.25 ft of standard black 1-in. pipe when the water was at a temperature of 63 F. The results of these tests are shown by line *AB*, Fig. 5. The experimental values shown by crosses were determined March 25, 1930, and those shown by triangles were determined March 26, 1930. On both days the water had a temperature of about 63 F and both sets of values lie quite accurately on the line *AB*.

After the tests with the orifices had been completed, the test with the unions only was repeated on April 4, 1930. At that time the water had a temperature of about 74 F and for this higher temperature a slightly lower friction head should exist because the viscosity of the water at 74 F is slightly lower than at 63 F. The values determined on that day are represented by the line *CD*. A separate study of these two lines (not reproduced in this paper) shows that the values derived from the line *CD* are about 7 per cent lower than those derived from the line *AB*. Theoretically, the difference should be only about 3 per cent, which is derived as follows: Reynold's numbers for a 1-in. pipe, and water at a velocity of 2 fps and at temperatures of 63 F and 74 F are, respectively, 15,490 and 17,040. From a curve showing the relation between Reynold's number and the friction factor in the expression $\left[f \frac{l}{d} \frac{V^2}{2g} \right]$ for a 1-in. black iron pipe, it appears that the values of the two friction factors are, respectively, 0.032 and 0.031, and that, consequently, the friction head should be about 1/32 or about 3 per cent smaller for the 74 F water than for the 63 F water. Since there may have been a slight error in the recorded water temperatures (which were taken in the upper portion of the supply tank) it was concluded that the method of measuring losses of head in this investigation is sufficiently accurate for the purpose for which it was performed.

To determine the friction head of one union, the line *EF*, Fig. 5, was drawn to represent the friction of 70 F water in a standard 1-in. pipe according to the equation $h = 0.006371V^{1.7767}$ taken from an earlier investigation by the author and published in University of Texas Bulletin No. 1759, page 31.

The differences between the friction heads shown by the lines *CD* and *EF* are the friction heads due to two unions. From these differences it appears that the friction head of one 1-in. standard black union is 60 milinches when the velocity of the water in the pipe is 10 ips and that, therefore, one union is equivalent, approximately, to 0.5 elbows, in the resulting loss of head.

The next seven series of tests were made with 0.027 in. copper plates,, having central circular orifices of 0.3 in., 0.4 in., 0.5 in., 0.6 in., 0.7 in., 0.8 in., and 0.9 in. in diameter; one plate being placed in each of the two unions. The re-

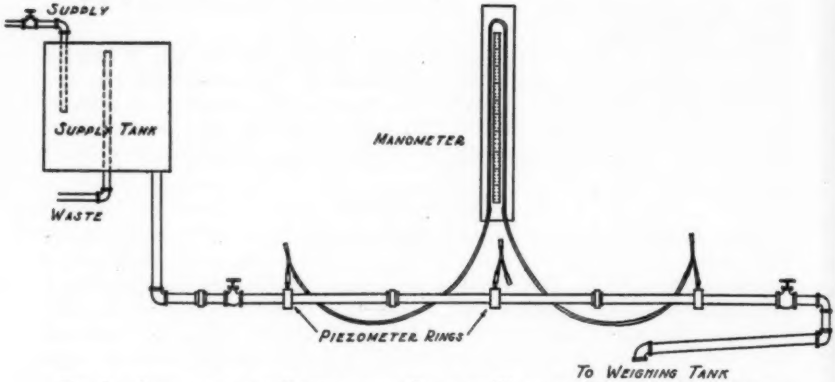

FIG 3. APPARATUS FOR DETERMINING LOSS OF HEAD IN DIAPHRAGM ORIFICES

sults of these tests were uniform except for the plates with 0.9 in. orifices. For this size orifice the friction head seemed to follow a law different from that according to which the friction heads in the other six plates varied. This difference may be caused by the fact that the 0.9 in. orifice differs so little in size from that of the pipe in which it was placed, and the results for this size orifice are not reported in this paper.[2]

The results secured with the remaining six sets of plates are shown by the lines *GH, IJ, KL, MN, OP,* and *RS* of Fig. 5. It is interesting to note how accurately the observed values fall on the several lines shown on the figure.

When these six lines were first drawn their positions and directions differed slightly from those shown in Fig. 5. It is evident, however, that the loss of head through these several orifices must follow some definite law; that is, the six lines must all either be parallel to each other or there must be a gradual and regular change in their inclinations; similarly, the values of the losses of head for a velocity of 1 in. (or for any other definite velocity) must vary according to some definite law. So, after the six lines had been drawn, a study was made of the several slopes and also of the several friction heads for a velocity of 1 ips and it was found that, by making very slight changes in the locations and in the slopes of some of the six lines, they could be drawn so that they would follow a definite law and still represent, as accurately as they had done before, the experimental values secured in the tests. The study referred to is not reproduced in this paper and the six lines shown in Fig. 5 are those which were determined after that study had been made.

The line *VW* in Fig. 5 shows the tangents of the slopes of the several lines and the line *XY* shows the value of the loss of head for a velocity of 1 ips, both lines being plotted against the diameter of the respective orifices.

Having determined these six lines with sufficient accuracy, the value of the

[2]After this paper had been published it was found that the clamping of the diaphragm orifices in the pipe unions had distorted the thin metal sheets to such an extent that the diameter of the 0.9 orifice, for example, was enlarged fully 1/16 in. This distortion may be responsible for the failure to secure consistent results with the larger orifices. The smaller orifices were not affected in this manner.

loss of head through one orifice, the 0.5 in., for example, can be easily found; it is half the difference between the values shown by the corresponding line, *KL* in this case, and the line *AB*. These differences can be read directly from the diagram of Fig. 5, or they can be determined with greater accuracy by calculations from the equations of the two lines.

This was done for all six cases and the resulting losses of head are shown by the corresponding lines of Fig. 6 in which the losses of head are expressed in milinches and plotted against the velocity of the water in the 1-in. pipe, expressed in inches per second. It is evident from the diagrams of this figure that if the velocity of the water in the 1-in. pipe is 2 ips, the loss of head caused by a 0.5 in. orifice is 150 milinches.

To make the diagram of Fig. 6 more useful, the five dotted lines were added to represent the losses of head caused by the five corresponding orifices, namely: 0.35 in., 0.45 in., 0.55 in., 0.65 in., and 0.75 in. The locations of these lines were determined graphically as shown in Fig. 8 by plotting the velocity of the water against the corresponding orifice for several losses of head, namely: 10 in., 3 in., 0.7 in., and 0.2 in., and then finding, from the resulting curves, the velocities corresponding to the desired orifices for the same losses of head.

Practical Application of Results

To explain the use of the diagram, let Fig. 7 represent a set of risers supplying three radiators in a gravity circulation heating system. Let each radiator supply 10,000 Btu and let each radiator have 1-in. connections and no valves. Let the friction head through each radiator and through its connections to risers, *i. e.*, from *A* to *B*, *or C* to *D*, or *E* to *F*, be 460 milinches. Let the temperatures of the water be 180 F and 160 F; the pressure head tending to force water through Radiator II will then be 996 milinches higher than that tending to force water through Radiator I.

If the two risers connecting those two radiators are of 1¼-in. pipe, the friction head in the 24 ft of 1¼-in. pipe will be 288 milinches; this is less than the available pressure head by 708 milinches, and it is necessary to provide 708 milinches of additional friction. From the pipe chart it will be found that if 10,000 Btu flow through a 1-in. pipe with a temperature drop of 20 F the velocity of the water will be 4½ ips in the 1-in. pipe.

Reference to Fig. 6 shows that for a velocity of 4½ ips and a friction head of 700 milinches, a 0.5 in. orifice must be used.

Similarly, the pressure head tending to force water through Radiator III is 1992 milinches higher than that tending to force it through Radiator I. If 1-in. pipe is used for the risers from Radiator II to Radiator III, the friction head in 24 ft of 1-in. pipe will be 312 milinches. This, added to the friction head (288

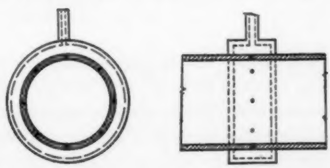

Fig. 4. Piezometer Ring and Eight Openings Through Pipe to Determine Pressure Heads

milinches) in the 1¼-in. risers, gives a total friction head of 600 milinches and that subtracted from the excess available pressure head (1992) leaves a balance of 1392 milinches, which must be provided to prevent Radiator III from robbing the two lower radiators of a portion of their share of the water.

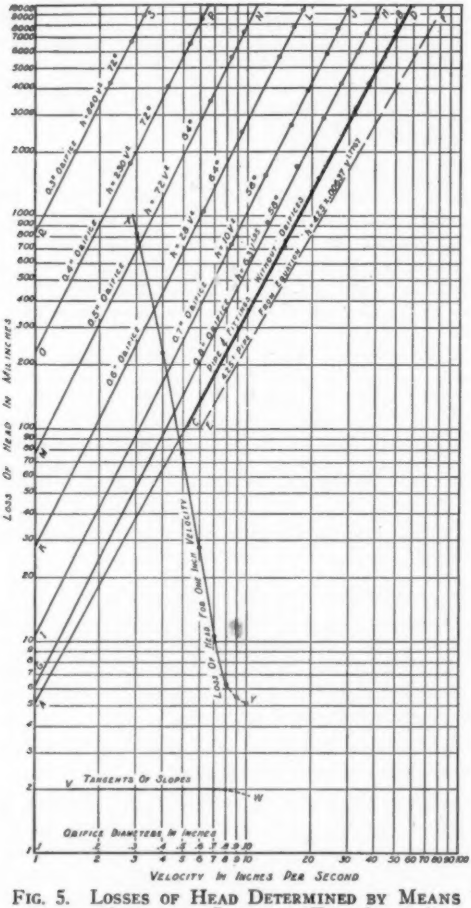

FIG. 5. LOSSES OF HEAD DETERMINED BY MEANS OF THE APPARATUS SHOWN IN FIGS. 3 AND 4

From Fig. 6 it appear that, for a velocity of 4½ ips and a friction head of 1392 milinches, a 0.45-in. orifice must be used in a union placed in the radiator connection. The experimental determinations described apply only to orifices in 1-in. pipe and the diagrams of Fig. 6 should be used only for orifices in 1-in. pipe.

To make this paper more nearly complete, and to check the accuracy of the results, Fig. 9 was added. The diagram of this figure shows the results secured with circular orifices in ¾-in. pipe in an investigation conducted by the Texas Engineering Experiment Station in cooperation with the A. S. H. V. E., and

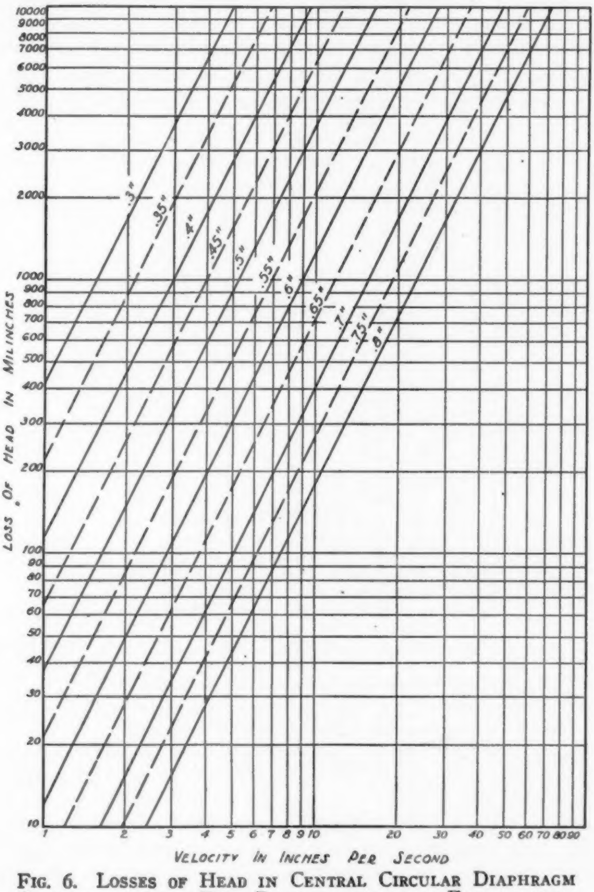

Fig. 6. Losses of Head in Central Circular Diaphragm Orifices in 1-in. Pipe Derived from the Experimental Values Shown in Fig. 5

reported in the paper Pipe Sizes for Hot Water Heating Systems by Giesecke and Smith, presented at the Summer Meeting of the Society in 1929.

In order to compare the results of the two investigations made at different times and by different methods, the line *AB* of Fig. 9 was drawn to represent the loss of head in a 0.3 in. orifice determined graphically from the values shown

in the diagram for ¼-in., 5/16-in., and ⅜-in., orifices by the method previously described and shown in Fig. 8.

A test was then performed to determine, experimentally, the loss of head in a 0.3 in. orifice in a ¾-in. pipe by the method used to determine the losses of head shown in Fig. 6 for orifices in 1-in. pipe; the results of this test fall so nearly on the line *AB* of Fig. 9 that it is impossible to draw a line through the points which differ materially from line *AB*. Since the losses of head for a 0.3 in.

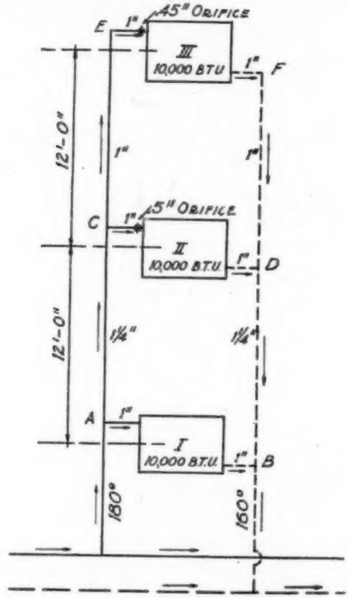

FIG. 7. A SECTION OF A HEATING SYSTEM TO ILLUSTRATE THE USE OF ORIFICES FOR INCREASING FRICTION WHERE NECESSARY

orifice determined by the two methods are practically identical, it is safe to conclude that both methods of determining losses of head in diaphragm orifices are reliable and that the results of both are sufficiently accurate for all practical purposes.

Finally, in order to compare the results of these investigations with those secured with circular diaphragm orifices in 4-in., 6-in., and 12-in. pipe and described in Engineering Experiment Station Bulletin No. 109 of the University of Illinois, the coefficients of discharge were calculated and the results recorded in Fig. 10. These coefficients were calculated as follows:

For a 1-in. pipe, a 0.5-in. orifice, and a loss of head of 6 in., the velocity of the

water in the pipe is 12.5 ips, and the quantity flowing through the orifice is 10.67 cu in. per second. The increase in velocity corresponding to a head of 6 in. is 68 in. and the resulting velocity is 12.5 plus 68 in., or 80.5 ips. The corresponding quantity of water which would flow through the orifice is 15.8 cu in. per second. The coefficient of discharge is $\frac{10.67}{15.8}$ or 0.674 as shown in Fig. 10.

A large number of these calculations were made and it was found that the coefficient of discharge is practically independent of the velocity of the water

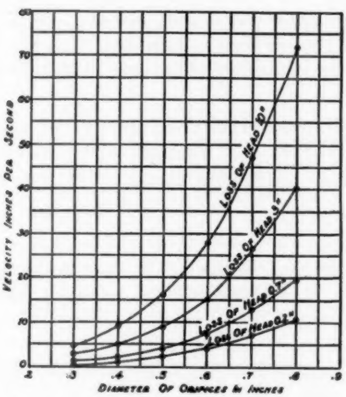

FIG. 8. A DIAGRAM TO BE USED FOR DETERMINING THE LOSSES OF HEAD FROM THOSE SHOWN BY THE FULL LINES OF FIG. 6

but varies with the size of the pipe and with the ratio of the diameter of the pipe to that of the orifice, as shown in Fig. 10.

The method adopted for these calculations is purely empirical and gives values which agree very closely with those quoted in Bulletin 109 of the University of Illinois referred to previously, when applied to the experimental data quoted in that bulletin. A method of calculating the coefficients of discharge which is theoretically correct cannot be applied in this case because the experimental data necessary for such calculations were not secured.

The variation with this ratio was found to be regular for orifices smaller than 0.8 in. when used in 1-in. pipe. The 0.8-in. orifice was evidently so large that its loss of head follows a different law from that governing the losses caused by the smaller orifices and this size should be excluded from a regular orifice table just as the 0.9-in. orifice was excluded in the earlier progress of this investigation.

A similar difficulty was experienced with the ⅜-in. orifice in the ¾-in. pipe.

The investigations at the University of Illinois indicate that it is difficult to secure precise measurements when the diameter of the orifice exceeds two-thirds the diameter of the pipe.

The results shown in Fig. 10 for 1-in. and ¾-in. pipe are so similar to those shown in the University of Illinois Bulletin No. 109 for 12-in., 6-in., and 4-in. pipe, and reproduced in Fig. 10 by dotted lines, that it seems possible to determine the loss of head for any central circular orifice in any pipe with considerable accuracy by interpolation between the experimental values of these two investigations.

CONCLUSIONS

The loss of head in submerged central circular diaphragm orifices in pipes varies materially with the size of the pipe and with the ratio of the diameter of the pipe to that of the orifice and only very slightly with the velocity of the water.

The losses can be determined very accurately and the resulting data used either to regulate the friction in a pipe line when the velocity is known or to determine the velocity and consequently, also, the quantity of water flowing through the orifice.

ACKNOWLEDGMENT

The experimental work was performed in the Mechanical Engineering Laboratory of Texas A. & M. College and the author is indebted to Prof. C. W.

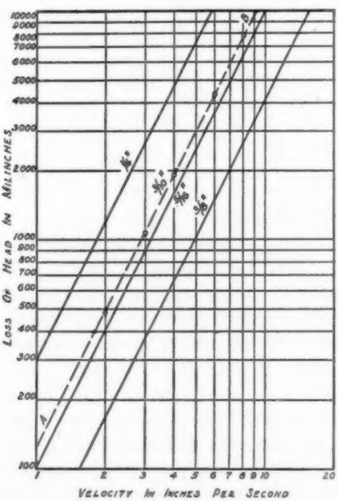

FIG. 9. LOSSES OF HEAD IN CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN ¾-IN. PIPE

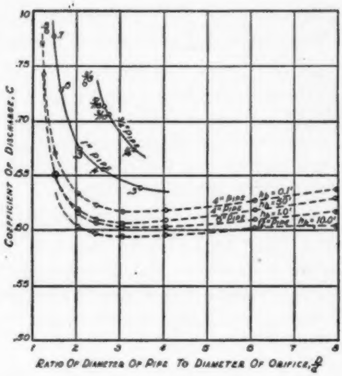

FIG. 10. A COMPARISON OF THE COEFFICIENTS OF DISCHARGE FOR CENTRAL CIRCULAR DIAPHRAGM ORIFICES IN ¾-IN. AND 1-IN. PIPE CALCULATED FROM THE VALUES SHOWN IN FIG. 9 AND FIG. 6 WITH THOSE FOR 4-IN., 6-IN. AND 12-IN. PIPE REPRODUCED FROM BULLETIN NO. 109, ENGINEERING EXPERIMENT STATION, UNIVERSITY OF ILLINOIS

Crawford and Prof. A. V. Brewer of the Department of Mechanical Engineering, and to Prof. T. A. Munson of the Department of Civil Engineering for valuable co-operation.

All drawings and practically all calculations and all laboratory work incident to this investigation are the work of W. H. Badgett, graduate student and research fellow at Texas A. & M. College, assisted by G. W. Lewis, a senior student, and much credit is due them for the accurate manner in which their work was performed.

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DISCUSSION

F. A. NAGLER (WRITTEN): Certain fundamental errors appear in the analysis on page 505 which lead one to distrust the reliability of Fig. 10. It does not appear to be a correct hydraulic procedure to add velocities in computing the value of the coefficient of discharge as Professor Giesecke has done on page 505. Hence, if the coefficients of discharge in Fig. 10 have been computed in the manner described, they are probably incorrect. The velocity head of the water in the pipe should be added to the head loss through the orifice, thus securing the velocity head corresponding to the speed of flow through the orifice. The writer thus obtains a velocity of approximately 69.2 fps through the half-inch orifice instead of 80.5 fps as obtained by the author in the example

given on page 505. Further computation will give a larger coefficient of discharge than that obtained by the author.

A. W. LUCK (WRITTEN): I can readily see the value of using orifices, but in these days when we have such keen competition and such a small margin of profit, we can hardly take the time to determine just what size orifice is necessary in order to secure certain results. In that case the charts presented will be very helpful and for that reason I am very much interested in this discussion and in the results presented.

I would like to suggest that a study be made to determine the relative height of a riser necessary to circulate water to a certain point below the mains or below the heater. I have a number of installations where the radiation is below the level of the foundation of the boiler, and there is no fixed rule of determining height of the riser in order to circulate water below the level of the boiler or return mains.

F. E. GIESECKE (WRITTEN): Replying to Professor Nagler, Figs. *A* and *B* show, respectively, the method of testing employed at the University of Illinois and at the A and M College of Texas. In the Illinois tests the total loss of head, h_a, and the maximum loss of pressure head, h_b, were determined and an expression

$$C = \frac{\left(\frac{D}{d}\right)^2}{\sqrt{\frac{2\,gh_b}{V^2} + 1}}$$

was developed by means of which the coefficient of discharge could be calculated. In the Texas tests, the loss of head, h_a, caused by the orifice, was the only quantity determined. The writer could think of no scientific method by which the

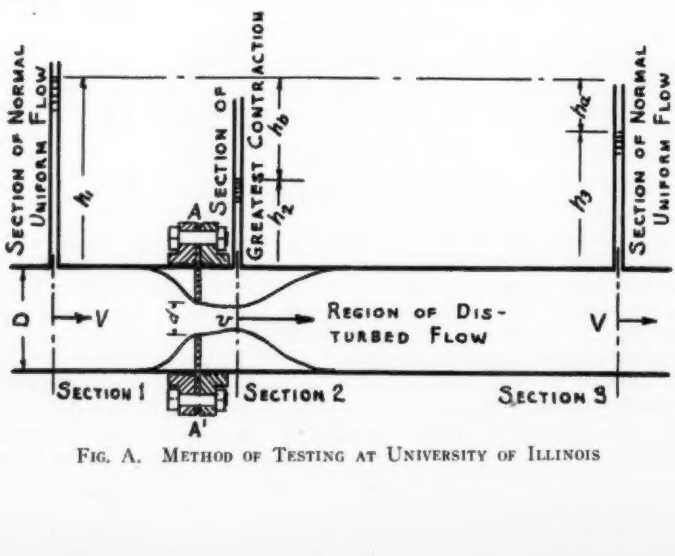

FIG. A. METHOD OF TESTING AT UNIVERSITY OF ILLINOIS

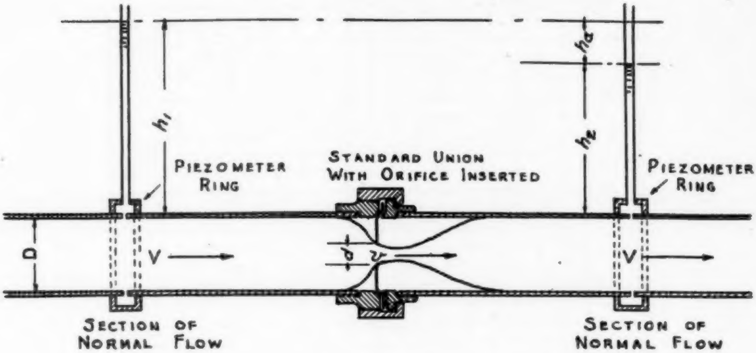

FIG. B. METHOD OF TESTING AT A. & M. COLLEGE OF TEXAS

coefficient of discharge could be calculated from the loss of head, so he adopted this empirical method.

By this method it was possible to calculate quite accurately the coefficient of discharge for the Illinois tests from the total loss of head. For four of the Illinois tests the average coefficient of discharge, calculated by the empirical method, differed by less than 2 per cent from the correct value. For example, for Illinois Test No. 195, the coefficient of discharge is 0.619; the value calculated by the empirical method is 0.626, or about 1 per cent too high. For the same test the value calculated by the method suggested by Professor Nagler is 0.68 or about 10 per cent too high. Since the empirical method gives results which are practically correct for the Illinois tests, the writer believes that the results calculated for the Texas tests and Fig. 10, which is based thereon, are sufficiently accurate for all practical purposes.

The only reason for presenting Fig. 10 was to show that the relation between the Illinois tests and the Texas tests is sufficiently close to justify interpolation between the two for intermediate pipe sizes.

Another study of this relationship is presented in Fig. *C*. The authors of the Illinois tests have shown that for any given velocity the loss of head is practically dependent only on the ratio of the diameter of the pipe to that of the orifice. The line *AB* shows the loss of head in terms of that ratio, for a velocity of 24 ips, for 4-in., 6-in., and 12-in. pipe according to the Illinois tests. The line *AC* shows similar data for 1-in. pipe, and the circles and crosses show, respectively, similar values for ¾-in. and 1¼-in. pipe according to the Texas tests. It is evident that in all cases the loss of head is a function of the ratio of diameters, as previously stated, and that the Texas values are somewhat lower than the Illinois values. This difference may be explained by assuming that the loss of head is a function not only of the ratio of diameters but also of the pipe diameter. It may be explained in part by the fact that for the Illinois tests the loss of head is the total loss which occurs between the points where the piezometer connections are made, whereas, in the Texas tests

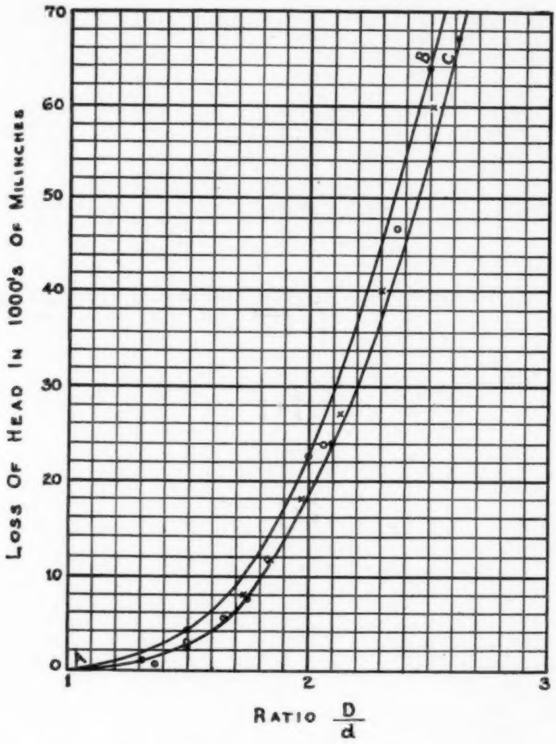

FIG. C. RELATION BETWEEN TESTS AT UNIVERSITY OF ILLINOIS AND A. & M. COLLEGE OF TEXAS

the loss of head is that for the orifice only. In the Texas tests one series of experiments was performed to determine the loss of head between the two piezometer connections when the union without the orifice plates was used and a second series when the orifice plate was inserted. By subtracting the losses of head found in the first series from those found in the second series the losses of head caused by the orifice only were found. They are the losses which are shown in Fig. *C*.

Replying to Mr. Luck, the writer believes it would be difficult to prepare rules by which pipe sizes may be determined when the radiator is placed below the heater or below the mains. Every individual case of such an installation can be easily designed. It is only necessary to select the pipe sizes so that the difference in weight of the water in the flow and return risers of the circuit will cause a pressure head equal to the friction head existing in the circuit when the water is flowing with the desired velocity.

No. 874

CARBON MONOXIDE CONCENTRATION IN GARAGES*

A. S. Langsdorf[1] and R. R. Tucker,[2] St. Louis

NON-MEMBERS

The results of cooperative research between the American Society of Heating and Ventilating Engineers and Washington University

THE original purpose of the investigation described in this report was to discover what relation, if any, exists between the concentration of carbon monoxide, gasoline vapors, and other noxious gases in any part of a garage, and those structural features of the building which may have a bearing upon these data. It was originally intended to investigate a number of garages representing a variety of types of heating and ventilating equipment, such as direct steam heating, steam heating plus forced ventilation, gravity warm air heating, warm air heating with fans, and unit type heaters. After the beginning of the heating season in October, 1929, it was found after repeated visits to the garages selected for observational purposes that because of the mildness of the St. Louis winter practically all of them operate for the most part without using the heating equipment, and in most cases with open doors and windows, except for an hour or two about midnight. During the winter there was a period in December and January of exceptionally cold weather, and this time was utilized to the fullest extent to collect samples of air in the Henri Chouteau Garage, 14th and Howard Streets. By the time this work had been completed the severe cold was followed by milder weather, precluding the possibility of additional work elsewhere, for the reasons cited.

The garage in which conditions were studied was the only one of the types selected which is ordinarily operated with closed doors. Its general design and arrangement are shown by the floor plan, Fig. 1, and by the photographs, Figs. 2, 3, and 4. It has a maximum length of 365 ft and a width of 110 ft, the volume of the building being 588,000 cu ft. It is used to house the maintenance and repair departments of a transcontinental bus line. The main heating equipment, of the warm-air type with blower, is located on the basement level in the southeast corner of the building, at the point marked *H* in Fig. 1. The intake takes

*This report covers one phase of an extended investigation of the subject of carbon monoxide concentrates in garages. It is proposed to carry on other phases later in a colder climate than St. Louis.

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Minneapolis, Minn., June, 1930.

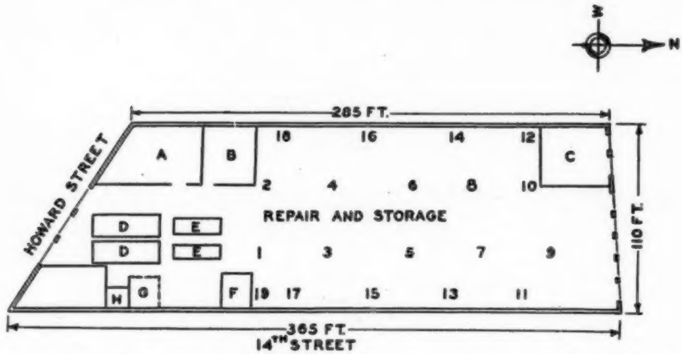

FIG. 1—HENRI CHOUTEAU GARAGE, 14TH AND HOWARD STS., ST. LOUIS. NUMBERS INDICATE SAMPLING STATIONS. *A*, MOTOR ROOM; *B*, STOCK ROOM; *C*, PAINT SHOP; *D*, WASH RACK; *E*, GREASE PIT; *F*, BLACKSMITH SHOP; *G*, FAN ROOM; AND *H*, HEATER

place through an opening in the floor guarded by the grating shown in Fig. 3, and the discharge is near the ceiling of the main floor. This heating unit has a capacity of 58,000 cu ft of air per minute, corresponding to a change of air every ten minutes. Doors having an area of 39 1/3 sq ft are available for the admission of outside air to the fan, but they were not in use during the period covered by these observations. An average temperature between 50 and 60 F is maintained at all times. An auxiliary heater of the extended-surface blast type is located in the northwest corner of the building to assist in the heating during unusually cold periods. The circulation of air in the building is in general along the longitudinal axis (south) through the main gallery and toward the southeast corner. This was checked by testing with a smoke cloud of ammonium chloride. In the vicinity of the pits for repair and inspection, the circulation is toward the center aisle.

The sampling stations, indicated by the numerals in Fig. 1, are about 50 ft

FIG. 2. EXTERIOR VIEW

apart, and are located along four longitudinal sections of the building. The stations along the east and west walls are staggered with respect to those in the main aisles. All samples were taken at a height of 4 ft above the floor, using bottles of two liters capacity. The water in the bottles was first saturated with the air to be sampled. The samples were analyzed for CO content by the iodine pentoxide method described by Fieldner, Henderson, Paul and Sayers in connection with the study of ventilation in the Vehicular Tunnels[3]. The apparatus for analyzing the samples is illustrated in Figs. 5 and 6.

Table 1 shows the average concentration at the stations indicated, throughout a month of observation, the samples having been taken between 2 and 3 o'clock

Table 1—Average CO Concentration

Station	CO Parts per 10,000
1	2.23
2	2.18
3	2.23
4	2.24
5	1.96
6	1.98
7	1.96
8	1.91
9	1.94
10	1.34
11	1.44
12	1.37
13	1.72
14	1.40
15	1.95
16	2.12
17	....
18	....
19	2.11
Fan Room	2.05
Office	2.71
Grease Pit	1.61

in the afternoons. The samples were not taken daily, but only on those days when the doors had been closed during the greater part of the day.

Samples were not taken at varying heights for the reason that the work of Randall and Leonhard[4] shows that where there is a marked movement of air the diffusion is complete. In their work there was no variation in CO content between samples taken at floor level and at a height of 5 ft above the floor.

The work of J. S. Haldane[5] in connection with the London Underground Railways shows that a CO concentration of one part in 10,000 is the maximum that should be permitted when there is to be exposure for a considerable time. Accordingly, it may be seen from the results indicated by Table 1 that considerable discomfort on the part of workmen in the building may be expected. A

[3]Ventilation of Vehicular Tunnels (Report of U. S. Bureau of Mines, Journal A. S. H. V. E., Jan.-Dec., 1926).

[4]Airation Studies of Garages, by W. C. Randall and L. W. Leonhard (Journal A. S. H. V. E., Oct., 1929).

[5]The Action of Carbonic Acid on Man, *Journal of Physiology*, Vol. 18, 1895.

FIG. 4—REPAIR AND STORAGE

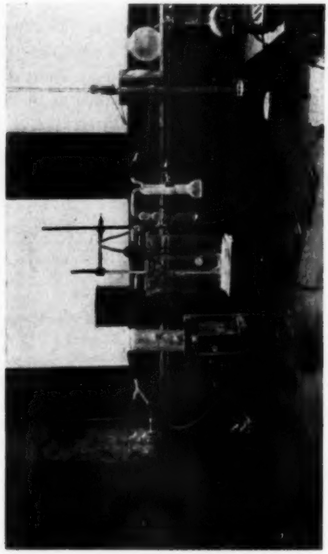

FIG. 5—GAS ANALYSIS APPARATUS

FIG. 3—FAN INTAKE

number of the men were questioned, and it was the general consensus that all suffered more or less, and that the effects were more noticeable when the temperature fell below the usual average. The majority of the men use aspirin to quiet the effects of the gas and reports of headaches and incipient nausea are common.

The roof of the main bay of the garage is fitted with a series of skylights and ventilators, but at no time during the sampling period were these ventilators open. There is no doubt that conditions would have been much improved had more outside air been circulated.

The average time required to take a sample was nearly 10 min, which is about the time to make a complete recirculation of the air in the building. Consequently, as samples are taken at different stations as the observer comes to them one after the other, the sample will represent different stages of the circulation cycle. Under steady conditions of operation this will not introduce serious discrepancies, but where conditions are changing rapidly, due to frequent car movements and opening and closing of doors, variations would occur. For strictly comparable results, the sample should be taken simultaneously and at frequent intervals, but this is hardly possible in a busy garage, even if there were enough observers to carry out such a program. Probably the best that can be expected is a record of average conditions at any given point.

FIG. 6—GAS ANALYSIS APPARATUS

The data presented in this report are not sufficiently extensive to justify final conclusions. It appears that the CO concentration increases as the fan intake is approached, and that elsewhere the concentrations in this particular building are higher than they should be.

DISCUSSION

A. C. WILLARD (WRITTEN): To the garage operator and mechanic, the concentration of carbon monoxide gas may become a matter of most vital importance, and if the conditions described in this paper are more or less typical, the problem merits much more serious consideration than it usually receives.

It is interesting to note that the average concentration was about 2 parts of

CO per 10,000 parts of air, which is much too high for continuous exposure. Moreover, the ventilation system failed to provide anything like uniform dilution of the monoxide gas, the concentrations varying by more than 100 per cent in different locations within the garage. Strangely enough, the higher *CO* concentration disclosed by this investigation occurred in the office. Incidentally, the office staff could endure such a condition better than the mechanics who are exercising and hence have a higher respiration rate than sedentary workers. Carbon monoxide absorption by the blood is augmented as the bodily activity is increased, hence workmen should be exposed to lower concentrations than non-workers for the same tolerance of *CO* in the blood, which is what really counts.

E. K. CAMPBELL: I do not think we want to take over the development of any instruments for measuring carbon monoxide. I am informed, however, that recording instruments have been developed and are in use in the Holland Tunnel. These instruments, although, they cost $1,000.00 apiece, are paying for themselves several times a year in cost of operating the plant. Of course, that plant is very elaborate and the cost of operation is great, so this scheme could not be applied to ordinary conditions. I would like very much to have the garage in St. Louis put in one of those instruments and control the ventilation accordingly, but the first thought that comes from Dean Langsdorf's report of his tests is that we have exactly the same condition there that they have in so many mechanical ventilation plants over the country, that they are not operated. The provisions for ventilation are not used. It is one of the things that we have tried to take into account in the development of the regulations for fire prevention.

Another thing that has been brought out very vividly by the experience in St. Louis, as Dean Langsdorf points out, is that these tests to be effective and to really let us carry on experimental data to learn what is needed to correct conditions when bad conditions are found, must be carried on in colder climates. So we are going to try to make arrangements with the Massachusetts Institute of Technology on some investigations in Boston. We do not know that we can do it yet but if they can be worked out that is probably where the tests will be made.

Mr. Davis, a member of our committee, and who has been connected with the Holland Tunnel both during its construction and its operation since, has written me in part as follows:

"Samples were taken 4 ft above the floor. As you know, carbon monoxide is lighter than air and therefore moves upward and in the case at hand the air currents are downward. In the experiments conducted by the Holland Tunnel Commission, under my supervision, we found that under similar conditions the carbon monoxide was passing the breathing zone of a man twice instead of once if the air currents were upward. In other words, the carbon monoxide would rise until encountered by a downward current when the carbon monoxide would be forced down again. I believe that is what happens in the case at hand and for that reason the samples should be taken at several heights."

We will plan to take that into consideration in future tests.

No. 875

CONTROL EQUIPMENT FOR GAS BURNING HEATING APPLIANCES

W. E. Stark[1], Cleveland, O.

MEMBER

THE American household has become mechanized in recent years. The domestic establishment has gone automatic. Pressing a button has relieved the housewife of many arduous duties, and has given the whole family automatic entertainment during the evening hours. And now, as a crowning achievement, automatic heat has appeared, leaving nothing else needed to make an evening in an American home wholly automatic, unless it be an automatic bridge player.

Those words *automatic heat,* although applicable to heating as it is practiced in the industrial or commercial establishment, no less than to heating as practiced in the domestic establishment, have their greatest significance when applied to home heating. They symbolize the emancipation of the man of the house, his wife, and his servants, from the drudgery of tending the fire, and from the occasional discomfort at times when the manually-controlled fire fails to keep pace with the vagaries of the weather. Nothing contributes in a greater degree to home comfort than attention-disdaining heating, and nothing contributes in a greater degree to the absent husband's peace of mind than the knowledge that the heating plant will continue to function in his absence.

Adaptability to automatic heating is not the exclusive prerogative of a single fuel. Gas, oil and coal are being successfully mechanized on a domestic scale. It is not the purpose of this paper, however, to draw any comparisons between these different fuels for obtaining automatic heating, but rather to survey, in a brief manner, the means available and utilized for harnessing one fuel, gas, and for making it one of the fuels adaptable to automatic heating as practiced in the modern mechanized home

Classification

Before considering the means by which control is secured, it may be well to consider the ends toward which those means are directed. The function of a controlling device as applied to a gas burning boiler or furnace is, in general,

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Minneapolis, Minn., June, 1930.

to regulate or to modify the rate at which the fuel is supplied to the main burners, in such a manner as either to maintain constant conditions, or to modify those conditions in accordance with a predetermined schedule. In order to accomplish that end the controlling means may be made responsive to certain independent variables, among which are:

1. Room temperature
2. Water temperature
3. Air discharge temperature
4. Outdoor temperature
5. Steam pressure
6. Gas pressure
7. Time
8. Pilot flame
9. Boiler water level

Certain of the controlling means responding to these variables may be designated as *operating* controls, some as *protective* controls, and some as both. The controlling means enumerated can be classified as follows:

Operating	*Protective*
Room Temperature	Water Temperature
Water Temperature	Air Discharge Temperature
Outdoor Temperature	Steam Pressure
Steam Pressure	Pilot Flame
Gas Pressure	Boiler Water Level
Time	

As a general rule, each control, or each function of control, is accomplished through the coordinate action of a controlling means and an actuating means. Controlling means in general usage today may be classified as:

(a) Mechanically operated
(b) Gas operated
(c) Electrically operated

DESCRIPTIONS OF CONTROLLING AND ACTUATING MEANS

The action of a controlling means is invariably that of a valve which shuts off completely the flow of gas to the main burners of the boiler or furnace, or modifies or *modulates* it, and the controlling means itself is simply a gas valve with a construction that lends itself to actuation by some variety of actuating means. Fig. 1 illustrates a controlling means of the mechanical type; Fig. 4, one of the gas-operated type; and Fig. 9, one of the electrical type.

The valve shown in Fig. 1 is essentially a globe valve, with a soft, gas-tight seat. It is opened and closed by the raising and lowering of a stem projecting through a stuffing box in the valve bonnet, and extending up into a case containing the various operating levers. By an ingenious arrangement of levers and pins, upward and downward movements of the outer extremity of the *operating lever* serve to open or close the valve gradually. Actuating means responsive either in changes in steam pressure or in water temperature can be adapted to produce this movement. (See Fig. 3.) The *shut-off lever,* to the outer extremity of which a weight is hung, closes the valve completely when room temperature demands are satisfied. To permit the valve to open, the

shut-off lever, with its attached weight, is held at the upper limit of its travel by a room-temperature controlled motor. Any standard make of room temperature thermostat capable of controlling a motor, can be used to raise and lower this lever. The motor can be attached to the lever by a chain, or it can be mounted directly on a bracket attached to the valve.

Fig. 2 shows how this type of controlling means can be actuated by boiler water level. A float chamber, serving also as a water column, is attached to the boiler above and below the water line. A float within the chamber, rising and falling with the water level, when in the position corresponding to a normal

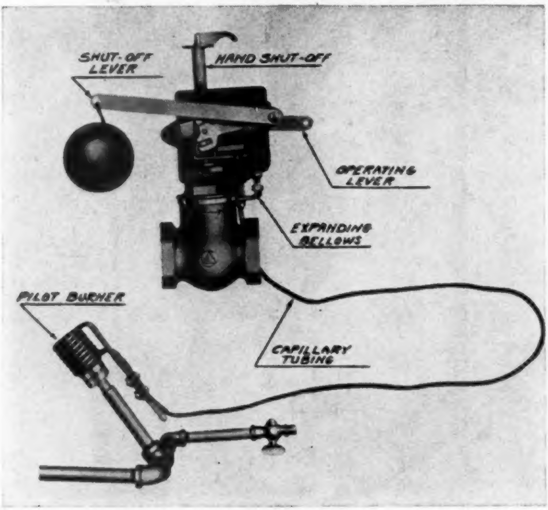

FIG. 1. MECHANICAL-TYPE GAS VALVE SHOWING THERMOSTATIC PILOT BURNER

water line permits a small crank supporting a weight to rest against a stop just beyond its vertical center. Dropping of the float, due to a subsidence in water level, causes the crank to be pushed through the small angle necessary to carry it away from the stop, and over center, whereupon the weight drops, depresses an extension of the valve stem, and closes the valve. On return of the water line to its normal level, the device may be reset, permitting the gas valve to be opened once more.

Fig. 1 shows the application of control by the pilot flame to this type of gas valve. A tube containing a volatile liquid is wrapped around a cup that surrounds the pilot flame. This tube is connected by a piece of capillary tubing to a bellows located within the compartment housing the valve-controlling levers. Pressure created in the tube and bellows by the presence of a flame

at the pilot, within the cup about which the tube is wrapped, causes the bellows to expand. This expansion is sufficient to compress a spring which will allow the main valve to open in response to a demand for gas. Absence of flame at the pilot, and therefore absence of pressure in the bellows, keeps the valve from opening regardless of the condition of steam pressure, water temperature, or room temperature control.

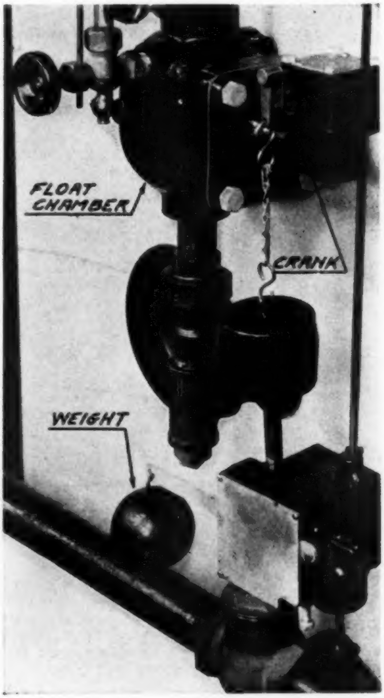

FIG. 2. MECHANICAL-TYPE LOW-WATER CONTROL

Fig. 3 shows a gas burning steam boiler provided with mechanical control and also showing the different elements pointed out.

Fig. 4 represents a gas valve of the gas-operated type. This valve consists essentially of a valve body containing a machined seat and suitable gas passages, to which is assembled a valve cover and a valve diaphragm which carries a valve disc. The valve contains in effect two chambers, separated by the diaphragm. The diaphragm is made of sheep-skin, extremely flexible, tanned by a process that makes it highly resistant to the action of the constituents of gas, and thoroughly impregnated with oil. The operation of the valve is due to differences produced in the pressures exerted on the upper and lower sides of the

diaphragm. With gas pressure beneath the diaphragm and atmospheric pressure above it, the diaphragm with its valve disc will be raised, allowing gas to pass through the valve. If, on the other hand, gas equal in pressure to that beneath the diaphragm and passing through the valve is allowed to fill the upper chamber, the total pressures above and below the diaphragm become equal and the diaphragm and valve disc fall due to their own weight, thus stopping the flow of gas. With the valve seated, the area exposed to gas pressure on the upper side of the diaphragm becomes much larger than that so exposed on

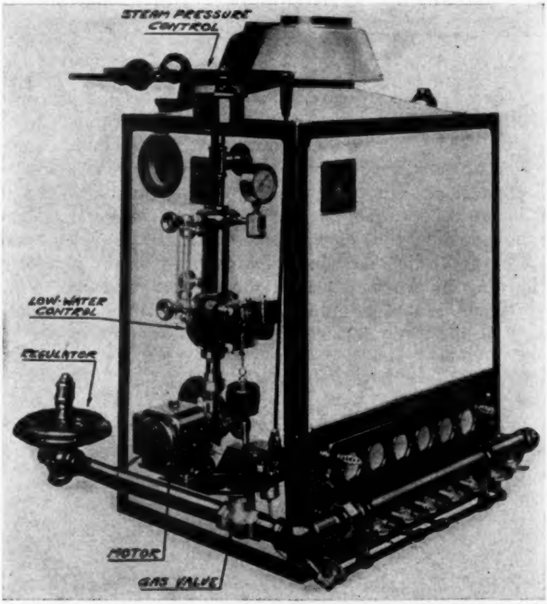

FIG. 3. STEAM BOILER WITH MECHANICAL CONTROL

the lower side, and the valve is held firmly seated by the preponderance of total pressure on the lower side.

There are two or three modifications of this construction in use, but the operating principles of the different types are practically identical. The construction shown in Fig. 4 is used in the major proportion of such valves.

To actuate this type of valve in response to changes in such conditions as boiler pressure, water temperature, water level, pilot flame, etc., it is only necessary that the changing conditions open or close a small valve which will admit gas to the upper chamber of the diaphragm valve when it is desired to close it. Conversely, the changing conditions must close this small valve when it is desired to have the diaphragm valve open. Fig. 5 shows an actuating means responsive to changes in steam pressure or water level. This device is connected to the steam boiler, above and below the water line and serves also

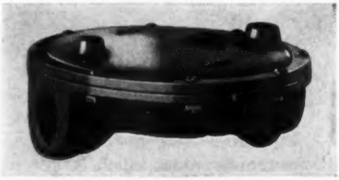

FIG. 4. GAS VALVE OF GAS-OPERATED TYPE

as the water column. The small valve at the top of this assembly is opened either by increasing steam pressure, which compresses the bellows against the resistance of the movable weight on the arm, or by receding water level which operates through the agency of the hollow float and suitable connecting parts. Fig. 6 shows an actuating means responsive to changes in water temperature. The unequal expansions of a copper tube immersed in the water, and an alloy rod enclosed in the copper tube, operate to open or close a small gas valve that is a part of the assembly, thus closing or opening the diaphragm valve. Fig. 7 shows a pilot burner which includes a feature that makes it necessary for the pilot to be burning in order to permit the diaphragm valve to open. A bi-metallic strip, when heated by the pilot flame, permits the small gas valve that is a part of the pilot burner assembly to remain closed. Cooling of

FIG. 5. GAS-OPERATED STEAM PRESSURE AND LOW WATER CONTROL

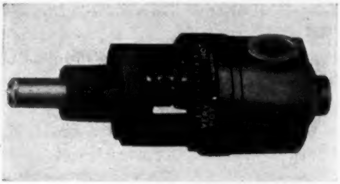

Fig. 6. Gas-Operated Water Temperature Control

this strip, due to accidental extinguishment of the pilot, opens the small valve, permitting gas to reach the upper chamber of the diaphragm valve, thus closing it. This feature prevents wasting of unburned gas in the remote event of a pilot being accidentally extinguished.

The device shown in Fig. 6 can with very few changes be used to control or limit the temperature of the air being discharged by a warm air furnace. The copper tube enclosing the alloy rod is inserted within the bonnet of the furnace and serves to prevent over-heating of the furnace in the event of circulation of air becoming restricted from any conceivable cause. For controlling the room temperature any of the usual varieties of room temperature thermostats can be used to actuate a small magnetic valve, which when open will permit gas to enter the upper chamber of the diaphragm valve.

The diaphragm valve can, if desired, be arranged so as to close gradually in response to increasing steam pressures, instead of closing completely when a predetermined maximum pressure is reached. On some classes of steam heating systems, this method of operation is advantageous. Fig. 8 shows a

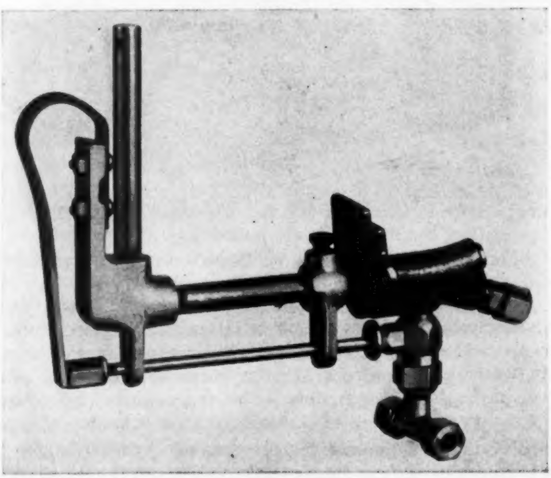

Fig. 7. Thermostatic Pilot Burner

gas-burning steam boiler provided with diaphragm valve control, with the different elements pointed out.

Electrical controls are, as their name implies, dependent on electrical current for their operation. The most usual type of electrical gas valve is shown in Fig. 9. It consists essentially of a globe valve body, with a rising stem carrying a rack which engages a pinion on the shaft of a small motor. Rotation of the motor, due to the completion of the operating circuit, draws the valve to the

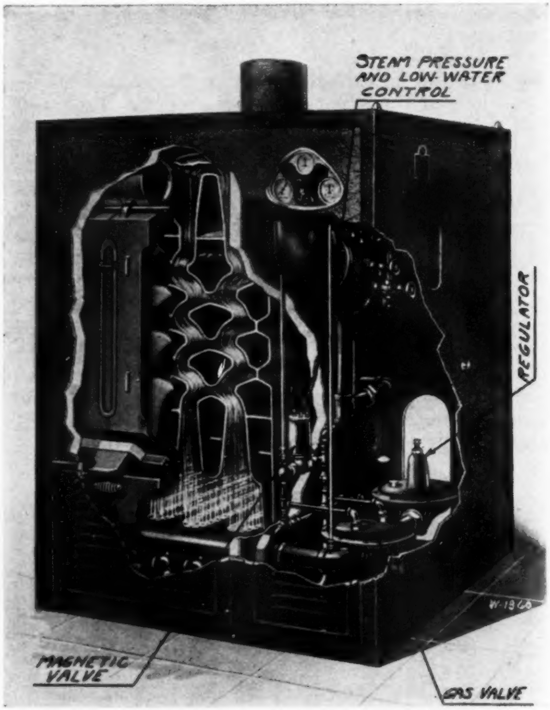

FIG. 8. STEAM BOILER WITH GAS-OPERATED CONTROL

upper limit of its travel, whereupon the motor stalls and holds the valve open so long as the electrical circuit is complete. Breaking of the circuit, which may be caused by the action of room or water temperature thermostats, steam pressure controls, water level controls, or pilot burner thermostats, will cause the valve to close through spring action. The thermostats and other actuating devices used for this purpose are of comparatively well known construction.

One piece of control equipment that is common to all completely equipped gas-burning heating devices is a gas pressure regulator, shown in Fig. 10. The purpose of the regulator is to maintain a constant gas pressure at the burner

orifices, regardless of variations in the pressure at which the gas reaches the premises. These variations in pressure are sometimes quite pronounced, particularly in the natural gas regions. Since there is one rate of gas supply at which the appliance will be delivering maximum efficiency and at the same time be burning the gas completely, it is essential that this one rate be maintained in spite of any fluctuations there may be in the street pressure. All gas burners are equipped with some form of metering orifice, so that by controlling the pressure on the inlet side of the orifice, the rate of gas flow may be controlled accurately.

The usual regulator is of the diaphragm type, with the diaphragm supporting a balanced valve. The pressure on the outlet side of the balanced valve (the regulated pressure) is conveyed to the lower side of the diaphragm. The diaphragm and valve are loaded by a spring, and regulation is accomplished by virtue of a balance of pressure being secured between the gas pressure on the lower side of the diaphragm and the spring pressure above the diaphragm.

Fig. 9. Electrically-Operated Gas Valve

By varying the initial compression of the spring, the outlet gas pressure may be adjusted, and once adjusted it will remain constant to within a very small percentage, regardless of inlet pressure variations.

Appliance and System Performance as Affected by Controls

The effect of a controlling device on appliance or system performance may be considered in two lights, (1) the effect on the comfort of those persons who are living in the atmosphere created by the heating system, and (2) the effect on the overall economy of the heating system, the overall economy including not only the bare cost of fuel but also the fixed charges on the investment and such service costs as are involved. On the one hand, there is an effect which is revealed only by the bodily sensations, and on the other, the price paid for that effect. As a general rule, so far as gas-burning heating appliances are concerned, the effect of a controlling method on fuel cost, that is, on appliance or system efficiency, is very small. After a gas-burning heating system is put under the primary control of a thermostat which maintains a substantially constant temperature within the heated space, the application of further controlling devices will produce very little change, in either direction, on the ratio of gas consumption to heating demand. The question thus becomes one of balancing the beneficial effect of a more or less complicated controlling arrangement against the fixed charges attributable to the first cost of the

arrangement and the added service, maintenance, and engineering costs. It is the writer's opinion that in many cases where attempts are made to correct faults in heating by the application of special controlling devices, the trouble might better have been attacked at the source, which is more often found in the heat distributing system than in the heat generating apparatus.

There is no need of discussing the effect of room temperature control on fuel economy. With an average heating season temperature of 35 F, and a desired indoor temperature of 70 F, each degree in excess of the desired 70 F means an addition of almost 3 per cent to the heat loss and therefore to the

Fig. 10. Gas Pressure Regulator

fuel consumption. A control, such as a room thermostat, maintaining a temperature that is in excess by a minimum amount of the lowest temperature compatible with bodily comfort, leads to maximum fuel economy. The question of whether clock control of room thermostats, whereby a reduced temperature is maintained during certain hours, results in increased fuel economy, has received a great deal of discussion. The writer has proved to his own satisfaction that in at least one type of residence construction, the maintenance of a reduced temperature during the night hours produces a definite fuel saving. This is by no means a universal and conclusive answer, however, because the possible creation of a peak demand on gas distributing systems during the heating-up period in the early morning, as well as the possible service calls chargeable to the clock, must also be considered.

It has already been pointed out that the automatically-controlled gas valve

may either shut off the flow of gas to the main burners completely, or shut it off only partially, by throttling or partially closing the gas valve in response to pressure or temperature changes. Arguments as to the effect on appliance efficiency of intermittent burner operation at full gas supply rate, against steady operation for longer periods at a reduced gas supply rate, are to no avail. Although a gas appliance operates at its maximum efficiency when it is burning gas at the rate for which it is designed, since at that rate the proper proportion of secondary air is drawn into the flame, the reduction in efficiency with a reasonable reduction in gas rate is so small as to be academic rather than real. The superiority, if any, of either method of gas control must be looked for in its effect on appliance operating characteristics and on the comfort of those living in the heated premises.

This paper has touched the subject of gas boiler and furnace controls in a

Fig. 11. Electrically-Operated Air Temperature Control

very sketchy manner. It has not attempted to present any of the numerous ramifications in control arrangements which may be resorted to in order to secure special effects. It is limited to a very general consideration of only those varieties of control which may be regarded as standard; and those standard varieties of control have demonstrated their adequacy to handle at least 99 per cent of all the gas appliance control problems encountered.

DISCUSSION

F. I. Raymond: The statement is made in this paper that the types of controls discussed have demonstrated their adequacy to handle at least 99 per cent of all the gas appliance control problems encountered. Although this statement is probably correct, I believe that improvements can be made and that there are worth-while controls outside of those listed in the paper.

Mention is made that outdoor temperature is one of the elements which can be used as an operating control. When outdoor temperature is used it has a tendency to even out the heating curve, to make the heat more continuous, instead of being a fluctuating load, hot radiators at one time and cold radiators at another time.

The room thermostat, of course, should be used in residential heating, but it has the limitations attendant upon time lag between the increase in radiator temperature and the increase in room temperature during which the radiators can build up an unnecessarily high temperature and a period of overheating follows. The period of overheating may not be objectionable, especially in the case of a very small building, but in larger buildings controlled by a thermostat alone the overheating can be quite pronounced due to the larger time lag. By using outdoor temperature as a steadying hand, as it were, over the room thermostat, the heating curve can be ironed out materially.

No. 876

ECONOMIC USE OF STEAM IN MODERN BUILDINGS

By F. A. Gunther,[1] Pittsburgh

MEMBER

Rapid progress has been made and is being made toward the simplification and perfection of the modern steam heating system, and yet with all the available highly efficient apparatus, very little has been accomplished in the reduction of overheating of modern buildings. The desirability of elimination of any wasteful practice is, of course, admitted. From the standpoint of cost, the building owner is much interested and from the standpoint of conservation of mineral resources there is again much favorable argument. Also from the standpoint of health, anything that can be done toward increasing resistance to colds and other respiratory diseases should certainly receive consideration.

The essentials for heating modern buildings must be kept clearly in mind before attempting to predict or state the magnitude of overheating of any building or group of buildings. Probably the three most important features of a satisfactory heating system are:

1. Proper distribution of the sources of heat within the space to be heated.
2. Sufficient and continuous supply of good air.
3. Regulation of temperatures within the comfort zone.

With these three essentials properly taken care of, much can be done toward properly and economically operating a heating system. The building operator must learn where the heat supplied to the building is consumed. He should be taught that any building is considerably like a sponge or sieve, soaking up heat and dissipating it through a myriad of heat leaks. Warm air is lost from the building through cracks around window sash and frames and through the walls themselves. Heat is conducted through the walls and glass and radiated from these wall and glass surfaces of the building. The chimney effect should be thoroughly explained to him, which should show him why more heat must be supplied to the lower floors of the tall building and less to the upper floors. In any existing building, very little can be done to reduce the conduction and radiation losses; but, on the other hand, large losses due to excessive

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Presented at the Semi-Annual Meeting of the American Society of Heating and Ventilating Engineers, Minneapolis, Minn., June 1930.

air change can be eliminated by proper caulking and weather-stripping of all openings.

With this much of the problem realized, the answer becomes tangible. Assuming that the construction of any particular building is known and the defects kept in mind, it becomes necessary to next find out what consumption of fuel should be expected to properly and economically heat the building. Much argument has resulted concerning the basis of comparison of heating consumption. Some engineers insist on comparing consumption on the basis of pounds of steam per cubic foot of volume. Others insist that the proper comparison should be on the basis of pounds of steam per square foot of equivalent direct radiation. The latter, in the writer's opinion, is preferable, especially when one realizes that in buildings of today the radiation has been designed to carry a temperature of 70 deg for a certain minimum outside temperature and wind condition, and that the calculations include the amount of heat necessary to take care of a specified number of air changes per hour. Therefore, when comparing heat consumptions on the basis of pounds of steam per square foot of equivalent direct radiation, the figures include not only radiation and conduction losses from the building, but also take into consideration the net volume of the building for the purpose of air change.

For many years, operators have been assuming that because any certain existing building consumes, for instance, 500 lb of steam per square foot of equivalent direct radiation that it is being heated properly, that there is not too much overheating and yet a figure as high as this may show as much as 50 per cent excessive consumption due entirely to this one cause. The figure may be entirely correct and basically economical for a warehouse, but for an office building with indirect or semi-indirect illumination and average occupancy, the heat from these two latter sources plus the other minor sources of heat in the building, exclusive of the heating system itself, this figure is usually nearly twice too high. When proper credit has been given for heat supplied by lights, occupants, domestic hot water systems, risers, and other miscellaneous minor sources of heat, this figure becomes less than 300 lb per square foot per season. This does not mean that any building now using more than 500 lb can be cut to 300 lb and all parts of the building properly heated, because there are innumerable exceptions. On the other hand, the writer made a survey and analysis of over eighteen buildings in Pittsburgh which do not control the supply of steam with any extra intention toward economy and has found that the average amount of steam used by these buildings was 572 lb per square foot per average season. Yet, a number of these buildings are now being heated for less than 350 lb and heated quite satisfactorily.

For the building operator to determine what the fuel or steam consumption for any existing building should be, it is necessary first to make a few simple tests. First, the cooling rate for the building should be determined by taking indoor temperatures over a period of time with no steam turned into the radiators of the building. This should be done for several out-door temperatures. These tests should be repeated at the same out-door temperatures with different wind velocities. Similar data should be taken to obtain the rate at which the building can be reheated after it has cooled. With these data available, it will be quite simple to plot curves between the coordinates of time and out-door temperature and correcting factors for wind velocity which will indicate at a glance

at what time steam may be shut off a building in the evening and should be turned on in the morning. One or two readings taken at specified hours during the night will serve as a check on these curves and allow proper corrections to be made for excessive drops in indoor temperature due to any change in out-door weather conditions. In addition, accurate data can be taken on the amount of fuel necessary to maintain average indoor temperatures by following the indications given by the results of these tests. Considerable of the wastage of steam during the night hours when the building is not occupied can be eliminated in this manner. A large amount of discussion has prevailed as to whether or not it is practical to save fuel at night, but any one who has

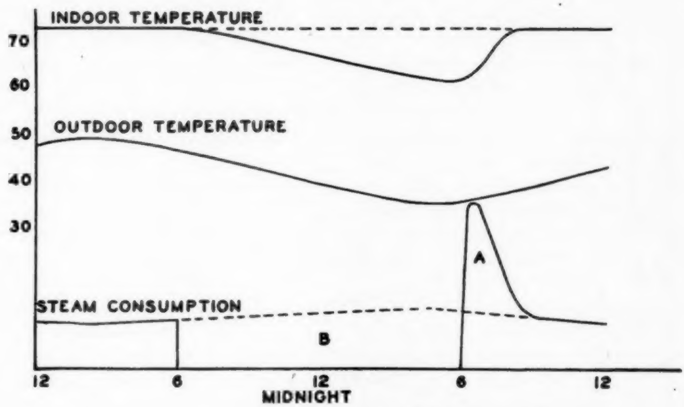

Fig. 1—Indoor Temperature and Steam Consumption for a Typical Building on an Average Day With Steam Off During the Night Hours

operated a building and made tests with and without shutting off steam during the night hours, knows that it is usually possible to save fuel by shutting off the supply of steam at night.

Fig. 1 illustrates theoretically what might be expected in the cooling of a building after the steam has been shut off on an average winter night. Indoor and out-door temperatures are both shown. The heating-up period for this day for an average building completely equipped with automatic temperature control requires about two hours. At the bottom is shown the steam consumption. Some idea of the proportionate savings by shutting off steam at night may be realized from comparison of area *A* with area *B*. Under the dotted line on the steam consumption portion of the chart is shown the amount of steam which would be consumed if the steam had not been shut off at night and the temperature had been maintained at the day-time level.

For the purpose of checking day consumptions the operator can plot the results obtained by dividing the daily steam consumption by the number of square feet of equivalent direct radiation in use in this quantity by a number of degree-days

for that particular day and plotting this amount against the number of degree-days per day. The resultant average curve drawn through these points should be hyperbolic and resemble that shown in Fig. 2. To properly regulate the temperature of a building during the hours of occupancy requires considerable attention on the part of the operator, yet it has been the writer's experience that the time spent is returned in dollars and cents many times over.

The supply of heat must at all times be adequate to keep the tenants satisfied, and stinting may result in extra work for the renting agent. On the other hand, a too liberal supply of heat is just as unsatisfactory and will result in a large number of open windows. Even without automatic control of tempera-

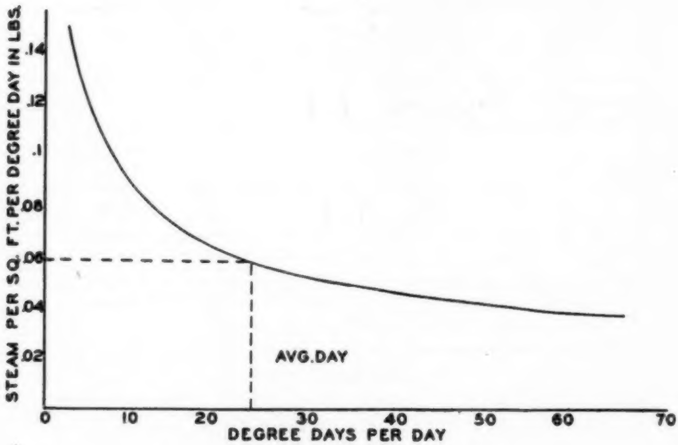

FIG. 2—TYPICAL HEATING RATE CURVE

tures both day and night it is possible, as previously stated, to reduce the fuel consumption to a degree. But from the vast amount of data now available, it appears that more can be accomplished by installing automatic temperature control equipment in the existing buildings and insisting upon its installation in the new buildings. There are a large number of automatic control devices manufactured today, each of which has its various points of advantage. These devices can be classified in two groups. *First,* those which control the temperature of the agent or medium of heat distribution and *second,* those which control the supply or quantity of the agent or medium. The second group can be further divided as follows:

1. Unit thermostatic control obtained through the use of one automatic thermostat controlling the supply to the entire building.

2. An amplification of (1) or what is known today as the automatically controlled zone system.

3. The more efficient result obtained from controlling groups of radiators.

4. The single radiator control.

The question has been often asked—What is good automatic temperature regulation? It is the maintenance of any specified temperature continuously, accurately, and automatically, and involves an automatic apparatus that supplies just the amount of heat necessary for the individual room it is controlling and no more. When this control is established throughout the building there will be consumed only that amount of fuel necessary to properly heat the building—any less is stinting; any more is wasteful. While this is a fundamental theorem, yet it is quite probable that it includes the whole truth about automatic temperature control.

The engineer of today designing buildings for tomorrow is neglecting his duty if he does not use every effort to include in his specifications the best type of automatic temperature control. Nor will he be quickly forgiven for this omission by the owner and future builder who are rapidly learning the great advantages to be obtained from a comparatively small addition to the initial cost of their buildings in the addition of this type of equipment.

Most of us have been guilty of installing radiation on courts and light-wells, assuming the same need for heat to be supplied to these locations as exists in the rooms with outside walls. Recent data show that the steam consumption of court radiation is approximately one-half that of outside radiation when calculated on the same basis. More information is needed to make a thorough determination along this line but certainly too much radiation has been used in the inside court rooms.

Engineers have been accustomed to figuring indirect radiation equivalent to direct radiation in ratios of from two to one, to seven to one, and yet the steam consumption of indirect systems rarely shows more than an average of one and one-half to one.

In the article entitled The Effect of Two Types of Cast-Iron Steam Radiators on Air Temperatures in Room Heating, by A. C. Willard and M. K. Fahnestock, which appeared in the March 1930 issue of *Heating, Piping and Air Conditioning*, the fallacy was pointed out of taking room temperatures 5 ft above the floor and yet nearly all heating specifications insist that temperatures be taken at this elevation.

Looking back over the past 15 years, considerable progress has been made in designing more efficient and more satisfactory systems of heating and ventilating. The next 15 years will, no doubt, bring forth even more improvements, and the writer predicts that the heating system of the near future will be not only more efficient, but will give greater satisfaction, will be entirely pleasing to the eye—rather more of a decoration, easy to keep clean and completely and automatically controlled. Positive and forced ventilating will be included as being as necessary as the supply of heat and the air will be really conditioned instead of just strained.

DISCUSSION

D. S. BOYDEN (WRITTEN): Exception should be taken to the statement in the first paragraph where it says "and yet with all the available highly efficient apparatus, very little has been accomplished in the reduction of over-heating of modern buildings." It seems to me that considerable has been done in the

last few years toward this end; for instance, the differential pressure system, the dual temperature control, balanced orifice, duo-stat system, the zone system or a combination of two or more of these systems. These systems are designed to eliminate waste in the use of steam, to furnish to the buildings in which they are installed only sufficient steam to make up for the heat loss.

The author states that "large losses due to excessive air change can be eliminated by proper caulking and weather stripping of all openings." The losses due to excessive air change have little to do with the economic use of steam, but they have considerable to do with the proper design and construction of the building in which the steam is to be used. A steam heating system in a building may work perfectly and still large amounts of steam may be wasted.

Quoting from the paper, "much argument has resulted concerning the basis of comparison of heating consumption. Some engineers insist on comparing consumption on the basis of pounds of steam per cubic foot of volume. Others insist that the proper comparison should be on the basis of pounds of steam per square foot of equivalent direct radiation. The latter in the writer's opinion is preferable."

It is not generally conceded that the better method is to compare on the basis of pounds of steam per square foot of equivalent direct radiation. It is the experience of district steam heating system operators, especially those operating large systems, that it is almost impossible to compare all buildings on the basis of square feet of equivalent direct radiation. On the other hand, the volume of the building is comparatively easy to obtain.

The amount of radiation to be installed in buildings is calculated by different methods. For this reason we have adopted the practice of using the size or volume of the building for comparison. The volume at least does not vary. To say that a certain building requires 500 lb of steam per square foot of radiation per season means nothing unless we know the rule by which the radiation was figured. In comparing it with another building it means nothing unless the radiation of both buildings is figured by the same rule or formula.

The term *equivalent direct radiation* might be acceptable if determined by a standard method, but a standard must also be used for every item that goes into the figuring of radiation. Manufacturers' rating of radiators do not agree within 10 or 15 per cent. Near the end of the paper the author says, "The question has often been asked—what is good automatic regulation? It is the maintenance of any specified temperature continuously, accurately and automatically and involves an automatic apparatus which supplies just the amount of heat necessary for the individual room it is controlling and no more." The automatic control should also be as simple as it is possible to make it. Experience has taught that complicated methods or systems of control will get out of order and when they do there is a tendency to allow them to remain in that condition.

The author's remarks about "figuring the ratio of indirect radiation equivalent to direct radiation" are interesting, but it would also be interesting to know why the steam consumption of indirect systems rarely shows more than an average of 1½ to 1.

A suggested method of comparing the steam consumption of buildings: calculate the heat loss in Btu per hour for each building according to A.S.H.V.E.

Guide standards. This is the heat loss after the building is heated and under stable operating conditions. Reduce to the number of thousands or millions of Btu. For example, if the calculated heat loss of a building is 500,000 Btu per hour, the base for this building would be $\frac{500,000}{1,000}$ or 500. Adopt a term for this factor (Thermerg). The annual steam consumption divided by this factor will result in pounds of steam per unit. All buildings on the same degree-day line may then be compared directly. For comparison of buildings under different degree-day conditions reduce to pounds per unit per degree-day, then compare.

G. W. Martin (Written): This paper is of value to the building manager and engineer supervising the operation of buildings. It also brings to the attention of the designing engineer matters which are too seldom given any thought after the building is in operation.

There appears to be some difference of opinion among engineers in New York as to whether the chimney effect in the so-called tower buildings will not be off-set by the increased exposure of such buildings. Observations are being made which will doubtless produce some interesting data.

The writer agrees with the author that the comparison on the basis of square feet of equivalent direct radiation is a more reliable one than that on a volumetric basis.

The writer recently made an investigation comparing the use of oil and coal for residence heating, and decided that the consumption of coal or oil per square foot of installed radiation would be the best basis of comparison. The houses surveyed were of the usual suburban type, very nearly identical in construction. Where the consumption of coal or oil per square foot was unusually low, the amount of installed radiation was usually excessive. Where the consumption of fuel per square foot was found to be higher than usual, the writer proved to his own satisfaction that the amount of installed radiation was inadequate.

It is obvious that an excessive amount of radiation divided into the pounds of fuel consumed would give a lower amount of fuel per square foot, while the smaller amount of radiation divided into the fuel consumption would give a higher amount per square foot. In the writer's opinion, therefore, a comparison of two buildings of the same size, construction and operating characteristics in the matter of service rendered, will determine whether or not a building is over or under-surfaced.

A schedule was prepared by W. J. Baldwin, Jr., of the New York Steam Corp., as a check on the economy of steam usage in the various types of buildings on the company's service. Mr. Baldwin assumed one square foot of direct radiation for each 100 cu ft of space heated. This, of course, would bring the comparison down to a volumetric basis. In some buildings, however, it will be found that the ratio will run up to one square foot of radiation for each 125 cu ft of volume. In these cases either the heating requirements are based on a temperature lower than 70 F or insufficient radiation has been installed. (The schedule referred to appears on page 378, the A.S.H.V.E. Guide 1931.)

The question of saving fuel by shutting off steam at night is no longer debatable. Even the average home owner knows that he can save fuel by using

some device to shut off the heat when he retires at night and turn it on again before he rises in the morning.

As to automatic temperature control, no modern building is complete without some application of thermostatic control of the heat supply. The heating plant in the modern building should also be arranged in zones so that on occasion different sections of the building may be heated as needed.

J. W. MEYER, JR. (WRITTEN): The methods adopted in conserving heat in buildings have been discussed before this and other societies and methods of procedure are fairly well established although not always practiced. The standard of comparison of the steam requirements of buildings is not so well established. The unit of comparison, either pounds of steam per square foot of required radiation or the pounds of steam per cubic foot of content of building, are frequently quoted as figures which might be expected to prevail under normal conditions. We, however, must differentiate between what a building should require, whichever unit of comparison is used, and the amount of steam necessary to satisfactorily service the building.

In setting up standard, as for instance 350 lb of steam per square foot per heating season per square foot of required radiation, we set up a figure which the building operator must attain, even though it may be at the sacrifice of quality of service.

Our present day distribution of space and the design of radiators does not always permit of a uniform distribution of heat throughout an entire building and it is practically impossible to design a heating system which will assure uniform temperature. We therefore resort to systems of control which appear to produce uniformity with a remarkable degree of success.

A heating system properly designed and controlled will condense a definite amount of steam per unit of radiation or per unit of content of building under normal temperature conditions and normal operation, which then becomes a standard for that building and any variations will be the result of changes in temperature conditions and in method of operation. Owing to the variations in the characteristics of different buildings, this unit figure will vary for different buildings and it would appear that the figure could be used only as a rough comparison of what might be expected.

The figure of 500 lb per square foot of radiation has been discarded for a lower figure, in some instances as low as 300 lb per square foot. The question in my mind is, whether we are not sacrificing quality of service in order to maintain a standard figure. In other words, while 300 lb per square foot may be a good figure to aim at, it may be attained only at the sacrifice of good service.

The determination of the type of temperature control to be installed involves several factors:

1. The steam consumption per unit of radiation or per unit of space heated.
2. The character of heat regulation required.
3. The cost of maintaining the control system.
4. The initial investment in temperature control.
5. The assurance of continuity of service of the temperature control equipment.

Obviously, the temperature control system to adopt is one which attains a minimum steam consumption consistent with adequate heating service; a heat regulation which will prevent under or over-heating; a minimum maintenance cost; a minimum justifiable initial cost; and a continuance of service of the control system under all conditions of load and temperature. In selecting a control system, proper recognition should be given to each of these factors. Low steam consumption is not the primary factor.

Except in the case of buildings purchasing steam, very little data on steam consumption are available, and the unit figures are therefore usually estimated on the basis of boiler efficiencies considerably higher than are found in actual practice. My experience indicates that estimates of fuel consumption of buildings are considerably in excess of the actual requirements.

S. S. SANFORD (WRITTEN): The author states that steam consumption of buildings should be compared on the basis of installed equivalent direct radiation. This method would be correct if the ratio of installed radiation to heat loss were constant for the buildings being compared. Unfortunately, this ratio varies considerably according to the extent to which an excess of radiation is provided. This is especially true in buildings heated by indirect radiation. Where heat loss data are not available, it has been the writer's experience that comparisons based on building volume give more consistent results than those based on installed radiation. Such comparisons should be limited to buildings in the same class and allowance should be made for the shape of the building. Another basis of comparison is steam used per square foot of exposed surface.

Fig. 2 would be less confusing if outdoor temperature were used in place of degree-days per day. The shape of the curve appears to indicate that steam is used wastefully in warm weather. This is because of the way in which degree-days are calculated. If the degree-days had been based on 70 F instead of 65 F, the curve would have been more nearly horizontal.

The degree-day should be based on the 24-hour average indoor temperature when a building is being operated with maximum economy. This will vary with outside temperature and will be different for different buildings depending on the temperature required during the day, required hours of heating and construction of the building. It is believed that in general the average temperature will be nearer 70 than 65 F. In the example given in Fig. 1, the average temperature appears to be nearly 70 F.

With regard to automatic temperature control equipment, it should be pointed out that any control operated by indoor temperature is affected by open windows and is subject to tampering by room occupants, and may therefore cause the steam consumption to be higher than necessary. It is believed that better economy can be obtained through fractional heating of radiators by the use of radiator orifices with the control in the hands of the operating force of the building.

In Fig. 1, the steam demand during the heating-up period is shown to be very high compared to that during the rest of the day. In the case of many buildings, the amount of steam required in the morning is much smaller than indicated. Figs. 3 and 4, which are records from two Detroit buildings, are presented to show that in some cases area A is almost non-existent.

MR. RAYMOND: In regard to Fig. 2, I think some exceptions should be taken

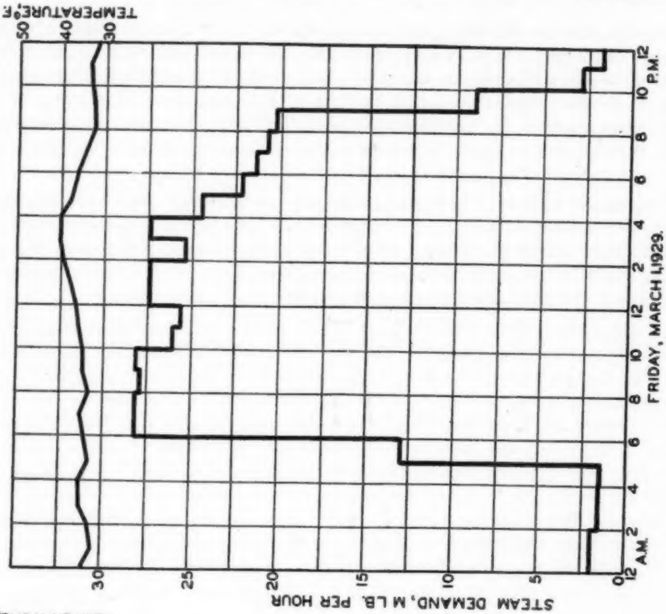

FIG. 4. STEAM HEATING LOAD CURVE FOR LARGE OFFICE BUILDINGS

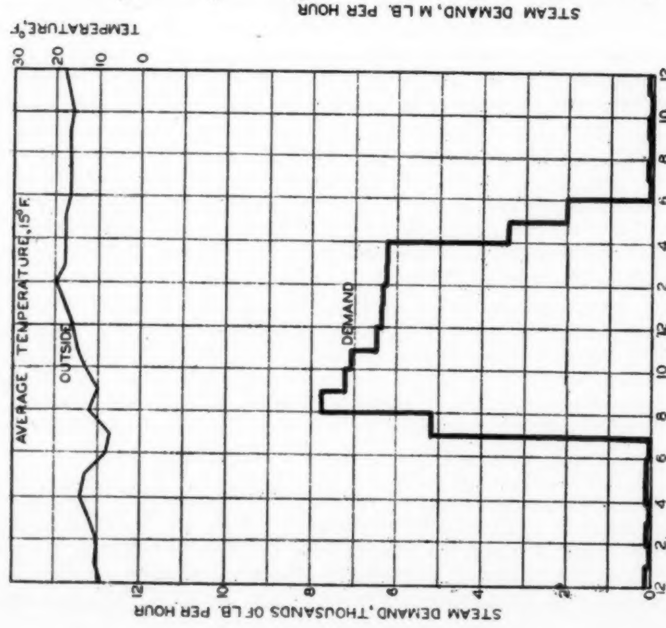

FIG. 3. LOAD CURVE FOR A DEPARTMENT STORE SHOWING ECONOMICAL OPERATION

to the curve shown. We find that approximately 3½ times as much heat per degree-day is required at weather about 65 F outdoors as is required around zero outdoors. I can see no reason why we should expect a higher proportion of heat per degree-day in mild weather than in cold weather. In fact, a lower per degree-day consumption would be expected in mild weather than in cold weather, because there is more sunshine and usually a lower wind velocity in mild weather than in cold weather.

The Indianapolis Power and Light Co. has made quite a study of the consumption per degree-day, and they have arrived at a rather unique and easy method of ascertaining whether or not a building is economically operated. They assume that very little heat, if any, is wasted in the most severe weather. By dividing the steam consumption of a building on a zero day, by 70, they get the steam consumption per degree-day. They figure then that if the building is economically operated they should get the same steam consumption per degree-day in mild weather as in cold weather. If it were possible to straighten out the curve of steam consumption per square foot per degree-day, we would effect an economy in mild weather. That, of course, is the only time we can effect a material reduction in steam consumption.

It seems to me that if we put this chart in the hands of a building operator and asked him to compare his consumption with this chart, we would tend to wasteful performance rather than to economical performance because the line theoretically should be a straight line or should taper off at the bottom.

There are a number of buildings in Indianapolis and Chicago in which the degree-day consumption is practically a straight line. In fact, in some buildings it is actually less per degree-day in mild weather than in cold weather.

MR. CAMPAU: In making comparisons of different classes of buildings at the Detroit Edison Co., we use the heated space, whereas I think most architects use the outside space, including penthouses and other outside parts of buildings.

One of our reasons for abandoning the use of radiation in making comparisons of steam consumption between several buildings and for using space instead is because we found that in buildings where there is a large proportion of indirect radiation, the indirect is likely to be used for a different number of hours than the direct radiation. Furthermore, if the speed of the fan is different than was intended when the computation for equivalent radiation was made, an error is introduced. In some cases, only a part of the installed indirect radiation is used, and in others, such as in some of the older theatres and churches, the fans are out of commission entirely and the direct radiation only is used.

At the beginning and the end of the season the buildings which have the largest consumption per degree-day, as compared with consumption during the cold weather, are those which have a large water heating load. Naturally, when a large quantity of water is heated, it adds to the consumption per degree-day.

F. D. MENSING: A step toward cubic content calculations by a heating engineer is a step back to our dark ages. In the past this system was commonly used and was known as the Mills System. No engineer today would use the Mills System, particularly so if he had ever had any experience with it.

A building that I have in mind was figured by the Btu method and required 4200 sq ft of steam radiation. The building was then designed to be insulated, the structure to all intents and purposes being the same in appearance and finish. It was again figured and 2500 ft of radiation was found sufficient.

It was finally constructed on the latter basis. Let us assume that two buildings had been built, one according to the original plan, and one according to the final plan; that is, one with 4200 and the other with 2500 sq ft. The cubic content are the same, but the amount of radiation varies by 1700 ft.

Another peculiarity in this particular building was that one room averaged 5 deg lower in temperature than the rest of the house. Investigation made by means of smoke bombs disclosed the trouble to be three high windows which leaked badly. Caulking overcame this difficulty and the room heated to the same temperature as the rest of the house. This house had a central thermostat controlling the heater. Now assume that 4200 ft had been installed in the insulated house. The fuel consumed due to this thermostat would have been approximately the same as if 2500 ft of radiation had been installed. This is obvious and requires no explanation and it is also just as obvious that there can be no actual relationship between cubic content and fuel consumption.

There is only one way to figure radiation, and that is as laid down in THE GUIDE. There is no one who is an engineer and has studied THE GUIDE who cannot figure almost any heating problem and who cannot get any comparison he wants even that of chimney effect in high buildings, etc. THE GUIDE is really a remarkable book. The trouble is that it is not used enough. It generally lays around and accumulates dust. It has been my experience that a lot of arguments that we have after the job is done could have been avoided if we had stuck more closely to the tested methods of figuring heat losses outlined in THE GUIDE. I would suggest a greater familiarity with THE GUIDE, particularly in checking up papers presented at our meeting.

I feel that Mr. Cassell can give us some enlightenment that may be very interesting on closing down plants entirely on certain periods and in trying to heat them up just before the building is used. We, as heating engineers, are dealing with a most delicate problem, the comfort of human beings. You can heat a building to 70 F, seat people in it and have them miserable so far as temperature goes. Mr. Cassell, I am sure, can give us some very interesting information on this subject.

J. D. CASSELL: I copied the idea of Mr. Waters of Chicago in laying out plenum systems, and I had no trouble whatever in heating our buildings with the plain plenum system (single duct running to each classroom), with the exception of Monday mornings after a very cold Saturday and Sunday. We shut the heat off on Friday afternoon and had no trouble bringing the temperature up to 70 F at the thermostat or the thermometer in connection with the thermostat. We would get 70 F whether on the inside wall or the outside wall, but I would get fifty reports of cold feet from various rooms. I overcame that by putting direct radiation under the windows opposite the air intakes, and particularly in the corner rooms. I did not have much trouble in the center rooms or just one-wall rooms, but where there were two walls to a room or a corner room, I invariably had trouble. We then copied the indirect system, reheating under each set of flues with indirect nests. We overcame somewhat this corner room defect by that system. I understand S. R. Lewis has a plan now whereby he can put in a plenum system by bringing three nests together and as one room goes up he shuts the heat off that one and then throws it all on two rooms and gets the heat from the three nests into one.

PRESIDENT HARDING: Load curves do not always go up, as temperatures go

down over comparatively short periods of time and do not always go down when the temperature goes up. From a theoretical standpoint it is a difficult problem to solve.

Not long ago I received a letter from an architect stating that he had figured the radiation for a certain building in accordance with THE GUIDE data. The amount calculated was installed and later it was found that a third of it could be removed. So I have a suspicion that the heat capacity and sun effect of the walls in that particular building had a considerable effect on the heating load.

F. A. GUNTHER: There is a lot of truth in Mr. Boyden's statement regarding the amount accomplished toward the reduction of overheating. However, I call his attention to the fact that with the exception of schools, comparatively few large buildings were equipped during their construction with temperature control systems during the last quarter century. The manufacturers have provided the highly efficient apparatus; it remains for the engineer to use it. I think his remarks and suggested method of comparing steam consumption of buildings are quite timely and should receive consideration by the Society. Messrs. Sanford, Raymond and Mensing make reference to Figs. 1 and 2 and wish to point out that these figures represent all of the steam consumption in any heating system including that in mains and risers. There are a number of systems and types of temperature control being used which shut off the supply of steam at the radiator only. The supply to the mains and buildings is often under manual control only. Under these conditions in mild weather, the loss from building risers and mains does very little toward increasing the temperature of the building and this is a constant loss as long as heat is turned on in the building. Obviously, if the consumption in the mains and risers of any building is 10 per cent of the total maximum consumption, then on a day on which there occurs only one or two degree-days, little, if any, steam will be consumed in the radiators, but the usual 10 per cent will be consumed in the risers. Therefore, the amount of steam consumed per degree-day is high in the mild weather. Further, assuming a constant consumption in the risers and mains of the building at zero degree-days, if heat is on in the building, even though no steam is used in the radiators themselves, the total steam consumption in pounds per degree-day is infinite but reduces very rapidly during the mild weather. It is somewhat more difficult to picture the results with some of the other economy apparatus now available, but the results closely approximate in the majority of cases, the curve shown in Fig. 2. This is partly due in some instances to the inability to hold constant and accurate temperature distribution throughout the entire building and steam saved by reducing its temperature or cutting off the supply to the mains and risers, is counterbalanced by the larger amount used due to the inability to hold the proper temperature distribution. Of course there are exceptions, the most important being a building which has uninsulated risers completely exposed to the space to be heated and these risers subject to automatic control. In this sort of combination, the curve begins to flatten out, but I have always found it higher in mild weather than average or cool weather.

In conclusion, if this paper and the discussion has caused only a very slight increase in interest of this subject, I feel that it has been worth while. I sincerely hope that much additional information on this subject will be forthcoming from the various members of the Society.

IN MEMORIAM

NAMES	JOINED THE SOCIETY	DIED
CHARLES D. ALLEN	1920	Jan. 1930
JOHN T. BRADLEY	1908	July 1930
MICHAEL J. CALLAHAN	1914	Oct. 1930
D. W. CHAPMAN	1914	Aug. 1930
JOHN M. CHASE	1915	Oct. 1930
HARRY MORTON DIX	1925	Sept. 1930
WILLIAM M. FOSTER	1914	April 1930
PATRICK GORMLY	1898	Dec. 1930
ALLEN HUBBARD	1919	Dec. 1930
RUFUS KAUFFMAN	1921	Dec. 1930
E. H. LOCKWOOD	1915	April 1930
J. J. MASON	1918	July 1930
H. BURTON MCLELLAND	1912	Jan. 1930
JACK B. MORRIS	1930	July 1930
SAMUEL C. PARTER	1907	July 1930
SAMUEL RAISLER	1921	July 1930
JOHN H. ROBERTS	1926	March 1930
WILLIAM A. RUSSELL	1918	June 1930
WILLIAM G. SNOW*	1903	Feb. 1930
JAMES B. WALKER	1919	Feb. 1930

*Past President.

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